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HEATING AND
AIR CONDITIONING

HEATING AND AIR CONDITIONING

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SIXTH EDITION
SIXTH IMPRESSION

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HEATING AND AIR CONDITIONING

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PREFACE

This book is intended as a textbook for use in engineering schools and by the individual who is studying the subject by himself. The method of treatment is therefore intended to be sufficiently clear so that a student who is presumably unfamiliar with the material will not be confused. Secondly, the attempt has been made to be sufficiently thorough so that the reader will acquire a solid theoretical groundwork and some knowledge of practice.

This sixth edition has been prepared in order to bring the book thoroughly up to date. The changes in the science of heating and air conditioning since the previous edition have not been revolutionary, and much of the previous text has therefore been retained.

An entirely new section on panel heating is included in the new edition. Also the principle of a reversed-cycle refrigeration system has received consideration from a theoretical and practical viewpoint. The chapter on Fuels and Boilers has been extensively revised, with the inclusion of flue-gas efficiency charts for coal, gas, or oil firing. Information relative to odor-removal apparatus and air sterilization has been added.

All the material on Gravity Warm-air Furnace Heating and Residence Air Conditioning has been completely revised to include the latest rating formulas and design procedures of the industry. Numerous changes will be found in the Hot Water Heating chapter, which incorporates more up-to-date information on design of small forced-circulation systems. Other recent developments in the field of heating, ventilating, and air conditioning have been adequately covered. In addition to the many new illustrations that have been added, several of the former cuts have been replaced with the latest design of equipment.

Chapter XX, which discusses in detail the design of an actual air conditioning system, was welcomed by many teachers as a new feature of the previous edition. That chapter has been completely rewritten in order to simplify the presentation and make the material more readily usable in the classroom. This aim has been accomplished by selecting a smaller and simpler design.

The material for this chapter was prepared by Mr. James N. Livermore, of The Detroit Edison Company, whose continuing cooperation is gratefully appreciated.

In revising the material on the thermodynamic properties of air- and water-vapor mixtures, there arose the important problem of whether this subject should again be treated in the simplified but not completely accurate manner in which it is usually handled or whether the more exact theory set forth by Goff and others should be used. In order to keep the book reasonably simple for the average student, the more simplified treatment has been continued in the text itself, while the more exact treatment has been given in the Appendix. It is probable that many teachers will wish to have their students become familiar with this appended material.

The work of preparing this revision has been done by Mr. John W. James, who becomes a coauthor. Mr. James has had an extraordinary background in teaching, in the preparation of engineering literature, and in the actual practice of engineering.

As heretofore, many individuals have directly or indirectly contributed material or suggestions, and manufacturers have been extremely generous in supplying illustrative material. Specific acknowledgment has been made wherever possible throughout the book.

J. H. W.

DETROIT, MICH.,
August, 1946.

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CHAPTER I

HEAT AND THE PROPERTIES OF STEAM AND AIR

1. Heat.—Heat has long been known to be a form of energy and is conceived to be a molecular motion or agitation. The intensity of the heat in a body is believed to be dependent upon the violence of this molecular disturbance. Every substance on the earth contains some heat, and to say that a body is "cold" means simply that it contains a relatively small amount of molecular motion.

Heat and many other forms of energy are mutually convertible. For example, heat energy is converted into electrical energy in a generating plant, and electrical energy is reconverted into heat energy in an electric stove. Heat energy is converted into mechanical energy in a steam locomotive, and some of this mechanical energy is reconverted into heat energy by the friction of the locomotive brakes.

2. Measurement of Heat.—In measuring heat there are two quantities to be considered, the *intensity* of heat and the *amount* of heat. A small piece of white-hot metal may not contain so great a quantity of heat as a pail of warm water, but the intensity of the heat in the former is much greater. The intensity of heat is denoted by the term **temperature.**

One measure of the intensity of heat in a body is its ability to transmit heat to a body of lower temperature. Heat will flow from a body of higher temperature to one of lower temperature but will never flow, of itself, from one body into another of higher temperature. When two bodies of different temperatures are placed in contact, a heat flow takes place until they are at the same temperature and thermal equilibrium is reached.

3. Measurement of Temperature.—The measurement of temperature is based upon an arbitrary scale that is standardized by comparison with some well-established physical fixed points. In mechanical engineering most measurements of temperature are made on the **Fahrenheit scale.**[1] On this scale the freezing point of water is taken at 32° and the boiling point at sea-level barometer at 212°, the tube of the thermometer between these points being divided into 180 equal parts, or degrees.

There is, however, some use of the **centigrade scale** among engineers. In the centigrade scale the distance between the freezing point and the boiling point is divided into 100 equal parts. The freezing point on the scale is marked 0°, and the boiling point is marked 100°.

If the temperature Fahrenheit is denoted by t_f and the temperature centigrade by t_c, then the **conversion from one scale to the other** may be made by the following equations:

$$t_f = \tfrac{9}{5}t_c + 32 \tag{1}$$

$$t_c = \tfrac{5}{9}(t_f - 32) \tag{2}$$

4. Absolute Temperature.—In any theoretical consideration of heat it is necessary to have some **absolute scale** of temperature. The point at which the **molecules of a substance would have no motion** is considered to be the **absolute-zero** point. This point is theoretically at 491.64° below the freezing point of water on the Fahrenheit scale, or 459.7° below the Fahrenheit zero. On the centrigrade scale the absolute zero is at $-273.1°$. To convert any temperature on the Fahrenheit or centigrade scale to absolute temperature the following formulas are used:

$$T_f = t_f + 459.7 \tag{3}$$

$$T_c = t_c + 273.1 \tag{4}$$

where the absolute temperatures on the Fahrenheit and centigrade scales are represented by T_f and T_c, respectively. The absolute-temperature centigrade scale is also known as the Kelvin scale, and temperatures on that scale are sometimes designated "degrees Kelvin."

5. Unit of Heat.—Heat must be measured by the **effect that it produces** upon some substance. The **unit of heat** used in mechanical engineering is the **heat required to raise the temperature of 1 lb. of water 1° Fahrenheit.** This is called the **British thermal unit** and is denoted by **B.t.u.** As this quantity is not exactly the same at all temperatures, it is necessary to specify further a definite temperature at which the unit is to be established. The **mean B.t.u.** is defined as $1/180$ part of the heat necessary to raise the temperature of 1 lb. of water from 32 to 212°F.

---

[1] The Fahrenheit scale is used throughout this book except where otherwise stated.

6. Specific Heat.—Specific heat may be defined as the **heat necessary to raise the temperature of a unit weight of a substance through 1°.** It represents the **specific thermal capacity** of a body. In English units the specific heat is the quantity of heat necessary to raise 1 lb. of a substance 1°F., expressed in B.t.u. Since the B.t.u. is the quantity of heat necessary to raise 1 lb. of water 1°F., the specific heat represents the **ratio** between the heat necessary to raise a unit weight of a substance 1° and the heat necessary to raise the same weight of water 1°.

When a substance is heated at **constant pressure,** its volume increases against that pressure, and **external work is done** as a consequence. The external work may be computed by multiplying the pressure by the change in volume. When a substance is heated at constant volume, no external work is done, as no movement is made against an external resistance. In any substance, such as a gas, that has a large coefficient of thermal expansion, the **specific heat of constant volume** will have a different value from the **specific heat of constant pressure,** the latter being the greater. The difference between the two specific heats in any particular gas must be equal to the heat equivalent of the external work done when a unit weight of the gas is raised 1° at a constant pressure.

The quantity of heat added to or removed from a body when its temperature is changed is equal to

$$WC(t_2 - t_1) \tag{5}$$

where W = weight of the body, lb.
C = specific heat of the material.
t_1 = lower temperature, deg. F.
t_2 = higher temperature, deg. F.

Values of heat capacity or specific heat per unit of *volume* for several building and insulating materials are given in Table 1. When such materials are subjected to variable heat flow involving intermittent heating and cooling cycles, these values are useful for determining equilibrium conditions.

TABLE 1.—SPECIFIC HEATS

Substance	Specific Heat	Substance	Specific Heat
Liquids:		*Metals:*	
Water	1.0000	Cast iron	0.1298
Alcohol	0.6220	Wrought iron	0.1138
Turpentine	0.4720	Soft steel	0.1165
Petroleum	0.4340	Copper	0.0951
Olive oil	0.3090	Brass	0.0939
Minerals:		Tin	0.0569
Coal	0.2777	Lead	0.0314
Marble	0.2159	Aluminum	0.2185
Chalk	0.2149	Zinc	0.0953
Stones generally	0.2100	Mercury	0.0333
Limestone	0.2170		

	Constant pressure	Constant volume		Constant pressure	Constant volume
Gases:			*Gases:*		
Air	0.2415	0.1729	Steam	0.5000	0.3500
Oxygen	0.2175	0.1550	Carbonic acid, CO_2	0.2479	0.1758
Hydrogen	3.4090	2.4122	Ammonia	0.5080	0.2990
Nitrogen	0.2438	0.1727			

	Specific heat, B.t.u. per lb. per °F.	Average density, lb. per cu. ft.	Heat capacity, B.t.u. per cu. ft. per °F.
Building materials:			
Brickwork	0.1950	100.0	19.5
Masonry	0.2000	150.0	30.0
Pine	0.4670	35.0	16.35
Oak	0.5700	43.0	24.51
Glass	0.1977	164.0	32.42
Insulating materials:[1]			
Glass wool	0.210	3.0	0.63
Rock wool	0.201	6.0	1.2
Corkboard	0.417	8.0	3.3
85 per cent magnesia	0.276	15.0	4.1
Corrugated asbestos paper	0.245	17.0	4.2
Wood fiberboard	0.341	14.5	5.0

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[1] From WILKES and WOOD, "The Specific Heat of Thermal Insulating Materials," *Trans. A.S.H. & V.E.*, 1942.

Example 1.—It is required to raise the temperature of a cast-iron radiator weighing 300 lb. from 70 to 212°. The temperature through which the iron would be raised would be 212° minus 70°, or 142°. The proper value obtained from Table 1 shows that to raise 1 lb. of cast iron 1° would require 0.1298 heat unit. To raise 1 lb. 142° would require 142 times 0.1298, or 18.43 heat units; and to raise 300 lb. 142° would require 300 times this amount or 5529 B.t.u., the heat required to heat the radiator.

Example 2.—Compare the amount of heat required to raise the average temperature from 70 to 150° of the insulation encasing a domestic heating boiler, assuming 15 cu. ft. of either glass wool or 85 per cent magnesia is needed. From Table 1, the heat capacity for glass wool is 0.63 B.t.u. per cu. ft. per °F. and for 85 per cent magnesia a comparable value is 4.1. With an average temperature difference of 80° the amount of heat required for the glass wool is 80 by 0.63 by 15 cu. ft. of material, or 756 B.t.u. For the magnesia insulation 80 by 4.1 by 15 equals 4920 B.t.u. Hence it requires $^{4920}/_{756}$ or about 6.5 times as much heat to reach equilibrium with the 85 per cent magnesia boiler covering as compared to an insulated jacket composed of glass wool, even though the conductivity of the two materials is practically the same.

7. Thermal Expansion.—Most substances undergo an increase in volume when heated and a decrease in volume when cooled. Gases are affected much more in this respect than liquids or solids, but the expansion and contraction of the latter are sufficiently great that in engineering work it is frequently necessary to take this property into account. For example, the pipes in a steam or hot-water heating system must be carefully designed so that the proper linear expansion will be provided for the necessary changes of temperature.

The change in length of a body for equal changes in temperature is approximately constant for moderate ranges of temperature, and the total change may therefore be calculated by the formula

$$e = CL(t_2 - t_1) \tag{6}$$

where e = change in length.
C = coefficient of linear expansion.
L = original length.
$t_2 - t_1$ = change of temperature.

If L and e are expressed in the same units of measurement, such as inches, then C is expressed in inches elongation per inch of original length per degree increase in temperature. The values of $C \times 10{,}000$ for a few materials are given in Table 2; values may be for all common materials found in engineering handbooks.

If a body is rigidly restrained so that it cannot expand or contract, a change in temperature will set up a stress equal to that required to lengthen or shorten it by the amount that it would have shortened or lengthened if unrestrained. Thus the stress created may be very great if expansion is not provided for. In a long steam pipe it is necessary to use a telescoping joint or other device to take up the linear expansion; otherwise a stress sufficient to rupture cast-iron fittings or permanently distort the pipe would probably be developed.

TABLE 2.—COEFFICIENTS OF THERMAL EXPANSION

	$C \times 10{,}000$ In., Change of Length per Inch of Original Length, per Degree Change of Temperature
Aluminum	0.128
Brass, cast	0.104
Cast iron	0.059
Copper	0.093
Nickel	0.071
Silver	0.107
Steel, Bessemer, soft rolled	0.063
Zinc	0.162

A liquid, like a solid, usually expands when heated and contracts when cooled. This property is utilized in the ordinary hot-water heating system, in which the circulation is produced by the greater density of the cool water leaving the radiators as compared with that of the hot water in the pipes supplying the radiators.

8. Heat Transmission; Radiation.—Heat may be transmitted from one body to other bodies or substances that are at a lower temperature in any of three ways: by **radiation,** by **conduction,** or by **convection.** Heat will not pass from one body to another of higher temperature without the application of some other form of energy. Usually more than one and frequently all three of these processes of heat transmission are involved to some extent in practical problems.

The radiation of heat through space is governed by the same physical laws as is the radiation of light. Heat is radiated in straight lines, and it is possible to have heat shadows. In fact, it has been proved that heat, light, and electromagnetic waves are identical in character, differing only in wave length.

The electromagnetic spectrum illustrated in Fig. 1 shows that the range of visible light is from about 4,000 (violet) to 8,000 (red) A.[1] Extending just beyond the visible portion of the spectrum are the infrared rays, which are simply heat rays and whose characteristic wave lengths are entirely dependent upon the temperature of the radiating body.

Many substances are transparent to heat rays, whereas others are opaque to them. Some substances are more nearly opaque to certain wave lengths than to others. Glass, for example, is more nearly opaque to the longer wave-length radiation from low-temperature sources than it is to the shorter wave-length radiation from high-temperature sources.

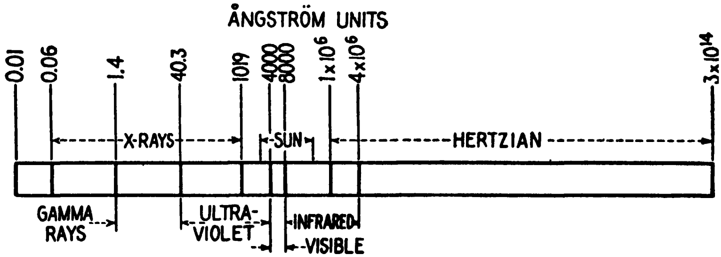

Fig. 1.—Electromagnetic spectrum.

The three principal factors that affect the amount of heat radiated from a surface are (1) the temperature of the surface, (2) the angle from which the surface is "seen" by the point in question to which heat is radiated, and (3) the physical nature of the surface. Stefan and Boltzmann showed that the amount of heat radiated from a body is a function of its absolute temperature.

The effect of the angle from which the surface is seen may be stated as follows: If heat is radiated from a point on a plane surface to a point in space at an angle ϕ from a line normal to the surface, the intensity of radiation toward that point will be equal to the intensity in the direction normal to the plane multiplied by $\cos \phi$.

The physical nature of the surface has a marked effect on the emission of heat by radiation. This property is called **emissivity** and is defined as the ratio of the emissive power of the surface to that of a perfect black surface. Polished surfaces and metallic paints have low emissivities as shown by the values for a few surfaces in Table 3 (from Hottel).[1]

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[1] One angstrom unit = 10^{-8} cm. = 3.937×10^{-9} in.

TABLE 3.—EMISSIVITY VALUES

Surface	Temperature, °F.	Emissivity
Lampblack	100	0.945
White enamel	73	0.906
Aluminum paint (10% Al)	212	0.52
Polished copper	176	0.018

In general, surfaces that have a high emissivity are poor reflectors. The **reflectivity** of a surface is the fraction of radiant energy incident upon it which is reflected, and the **absorptivity** is that fraction which is absorbed. The absorptivity is numerically equal to the emissivity for any given surface.

Under ordinary conditions a surface radiating heat is also receiving heat from other sources. The net amount of heat radiated from a surface to another at a lower temperature is expressed according to Stefan's law, as follows:

$$H_R = AK(T_1^4 - T_2^4) \qquad (7)$$

where H_R is the net quantity of heat radiated from one surface to the other, A is the area of one of the surfaces, T_1, and T_2 are their respective absolute temperatures, and K is a constant that takes into account the average angle at which one surface "sees" the other and the emissivities of the two surfaces. It is beyond the scope of this book to take up in detail the complex problem of the application of Stefan's law, but a knowledge of the general principles involved is often useful.

Irrespective of the method of heating utilized for maintaining desired temperatures within an enclosure, all the interior surfaces radiate and reradiate from hot to cold surfaces until equilibrium temperatures approximately prevail. In most structures the conductivity of the materials that form the boundary surfaces, such as floors, walls, ceiling, doors, and windows, varies considerably, and for that reason the respective surface temperatures will be different. The **mean radiant temperature** of all these surrounding surfaces may be defined as a uniform temperature at which a black surface would radiate with an intensity equal to the mean observed. A technique for measuring mean radiant temperatures will be found in Chap. V, together with a discussion of various methods of producing panel or radiant heating.

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[1] The emissivities of certain surfaces vary greatly at different temperatures. For such surfaces there is no similarity between the absorptivities of high-temperature (short-wave-length) solar radiation and the emissivities at ordinary temperatures. The absorptivities of various surfaces for solar radiation are given in Chap. XIX.

9. Conduction.—Heat will pass from a body to another body at a lower temperature which is brought into contact with it, and if one part of a body is at a higher temperature than another part, there will be a flow of heat through the body. The transmission of heat in this manner is known as **conduction.** A familiar example of this phenomenon is the flow of heat along an iron bar, one end of which is heated in a fire.

The ability of different materials to conduct heat differs considerably. Metals are the best conductors of heat, while such materials as wood, felt, and asbestos are very poor conductors and consequently are good heat **insulators.**

The **conductivity** of a material is the amount of heat that would be conducted through a plate of the material of unit area and unit thickness with a unit difference in temperature between the two sides of the plate.

The total amount of heat transmitted per unit of time through a plate or wall will depend upon the material, upon the difference in temperature of its two sides, and upon its thickness. This may be expressed as follows:

Let

k = the conductivity of the material, B.t.u. per hr. per sq. ft. of area per in. of thickness per deg. difference in temperature.

t_1 = the temperature of the warmer side of the plate, deg. F.

t_2 = the temperature of the cooler side of the plate, deg. F.

A = the area of surface, sq. ft.

x = the thickness of plate, in.

H_c = the total quantity of heat transmitted, B.t.u. per hr.

Then

$$H_c = \frac{Ak(t_1 - t_2)}{x} \tag{8}$$

The **conductivity of the heat path** is then Ak/x, and the **resistance** of the heat path is its reciprocal x/Ak.

Example.—Suppose a boiler plate 5 ft. square and ½ in. thick to have a temperature of 70° on one side and 200° on the other side. Assume the conductivity of the metal to be 240 B.t.u. per hr. per sq. ft. of area per inch in thickness per degree difference in temperature. The total heat transmitted per hour is then

$$H_c = \frac{25 \times 240(200 - 70)}{\frac{1}{2}} = 1{,}560{,}000 \text{ B.t.u. per hr.}$$

The conductivity k in Eq. (8) is assumed to be constant throughout the thickness of the plate. Actually for most materials k varies with the temperature of the material, in some cases increasing and in others decreasing with decreasing temperature. In the transmission of heat through building materials, the range of temperature through the thickness of the material is relatively small; and if k is regarded as the average conductivity at the existing temperatures, the error will be negligible.

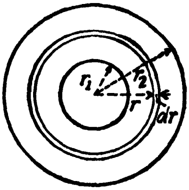

FIG. 2.

Another case frequently encountered is that of radial heat flow in a cylinder, such as the flow through the heat-insulating material on a steam pipe.

In Fig. 2 let

r = variable radius between the limits r_1 and r_2.
dr = thickness of very thin cylinder of radius r.
dt = temperature difference between the inside and outside of cylinder wall dr.
H_c = heat flow per unit length.
k = conductivity.
t_1 = temperature at the inner surface.
t_2 = temperature at the outer surface.

$$H_c = \frac{k\,2\pi r\,dt}{12dr}$$

$$H_c = \frac{2\pi k(t_1 - t_2)}{12\log_e \frac{r_2}{r_1}} \tag{9}$$

If r_1 and r_2 are expressed in inches, H_c will be in B.t.u. per hour per foot of length, k being expressed as in Eq. (8).

10. Convection.—When heat is transmitted by radiation or conduction, no motion of the substances (other than molecular activity) is involved. In the case of convection, however, the essential process is the flow of a fluid (liquid or gas) over a solid surface, accompanied by a transfer of heat between that surface and the fluid. The movement of the fluid may be due to changes in its density caused by the change in its temperature, giving rise to so-called **natural convection**; or the movement may be created by mechanical means, in which case the process is termed **forced convection.** It should be noted that the transfer of heat at the interface of the fluid and the solid takes place by conduction. The essential characteristic of the process of convection is the continuous renewal of the fluid layer in contact with the surface.

The transfer of heat by convection is a complex phenomenon in that it is affected by a large number of variable factors, such as the nature of the surface and the velocity of the fluid over the surface. Increasing the velocity of the fluid causes a more frequent renewal of the layer of fluid in contact with the surface and increases the rate of heat transfer.

11. First Law of Thermodynamics.—**When mechanical energy is produced from heat, a definite quantity of heat is used up for every unit of work done**; and, conversely, when heat is produced by the expenditure of mechanical energy, the same definite quantity of heat is produced for every unit of work spent. This first law of thermodynamics might also be called the law of the conservation of energy. The relation between work and heat has recently been determined with great accuracy, and the results show that **1 B.t.u. is equivalent to 778.26 ft.-lb.** This factor is called the **mechanical equivalent of heat,** or Joule's equivalent.

STEAM

12. The Formation of Steam.—Many substances can exist in more than one state under the proper conditions of temperature and pressure. Water exists as ice at low temperatures and as steam at higher temperatures, the temperatures depending upon the pressure. If heat is applied to a vessel partly filled with cold water, the temperature of the water will rise until a certain temperature is reached, at which small particles of water are changed into steam. The water is then said to **boil.** The temperature at which the water boils depends upon the pressure in the vessel. If the pressure is raised, as by partly closing the outlet, and if the addition of heat continues, the temperature of the water will rise to the point corresponding to the existing pressure.

Steam, when still in contact and in equilibrium with the water from which it is produced, remains at the temperature corresponding to its pressure. Under this condition the steam is said to be **saturated.** If it is removed from contact with the water and further heated, its temperatures will rise and the steam will then be **superheated.**

13. Superheated Steam.—Superheated steam is steam at a temperature higher than the temperature of the boiling point corresponding to the pressure. If water is intimately mixed with superheated steam, some of the heat in the steam is used in evaporating the water and the temperature of the steam is lowered. If sufficient water is added, the superheat would be entirely used up in evaporating the water and the steam would then be saturated. Superheated steam can have any temperature higher than that of the boiling point. When raised to any temperature considerably above the boiling point, it follows fairly closely the laws of a perfect gas.

14. Saturated Steam.—When steam is at the temperature of the boiling point corresponding to its pressure, it is said to be **saturated.** If this saturated steam contains no suspended moisture, it is said to be **dry saturated steam**; or, in other words, **dry saturated steam is steam at the temperature of the boiling point and containing no water in suspension.** If heat is added to dry saturated steam, not in the presence of water, it will become **superheated.** If heat is taken away from dry saturated steam, it will become **wet steam.**

Steam at a pressure equal to or slightly above atmosphere is commonly known as **vapor.** It should be remembered, however, that **the difference between vapor and steam is merely one of pressure** and that vapor is in no sense a different state of the substance.

Dry saturated steam is not a perfect gas, and the relations of its pressure, volume, and temperature do not follow any simple law but have been determined by experiment.

15. Properties of Steam.—The heat used in the formation of 1 lb. of superheated steam at any pressure from water at 32° may be divided into three parts: (*a*) The **heat of the liquid,** which is the heat required to raise the temperature of the water from 32° to the temperature of the boiling point; (*b*) the **latent heat of evaporation,** which is the amount required to change 1 lb. of water at the temperature of the boiling point to dry saturated steam at the same temperature; and (*c*) the **superheat,** which is the heat added to 1 lb. of steam to raise it from the boiling-point temperature to the final temperature.

16. Heat of the Liquid.—The heat of the liquid may be determined for any boiling-point temperature by the expression

$$h_f = c(t - 32) \tag{10}$$

where h_f = the heat of the liquid.
t = the boiling-point temperature.
c = the specific heat of water.

For approximate results c may be taken as equal to 1.

The change in the volume of the water during the increase in temperature is extremely small; the amount of external work done may usually be neglected, and all the heat of the liquid may be considered as going to increase the heat energy of the water.

The heat of the liquid, together with the other properties of saturated steam, is given in Table 4 for various steam pressures.

17. Latent Heat.—The latent heat of steam, or the **heat of evaporation,** has been defined as the **heat required to convert 1 lb. of water at the temperature of the boiling point into dry saturated steam at the same temperature.** Experiments show that the latent heat, designated by h_{fg}, diminishes as the pressure increases.

When water is changed into steam, the volume is greatly increased, so that a considerable portion of the latent heat is used in doing external work. The remainder may be considered as being utilized in changing the physical state of the water. Let p be the pressure at which the steam is generated, v_g the volume of 1 lb. of steam, and v_f the volume of 1 lb. of water; then the external work done is equal to

$$p(v_g - v_f) = pv_{fg} \tag{11}$$

At 212° the external work done in producing 1 lb. of steam is equivalent to 73 B.t.u., or about one-thirteenth of the latent heat.

When steam condenses, the same amount of heat is given up as was required to produce it. In the steam heating system the latent heat is added to the water in the boiler, converting it into steam. The steam is conducted to the radiators, in which it condenses, and in condensing it gives up its latent heat, which goes to warm the room.

18. Total Heat or Enthalpy of Steam.—The **total heat or enthalpy of dry saturated steam** is the heat required to change 1 lb. of water at 32° into dry saturated steam. This quantity will be designated by h_g, and

$$h_g = h_f + h_{fg} \tag{12}$$

The total heat in 1 lb. of steam under any condition of moisture or superheat is the amount of heat required to change it from water at 32° to its existing condition.

When steam contains entrained water, the percentage by weight of dry steam in the mixture is termed the **quality of the steam.** If q represents the quality of the steam, then the latent heat in 1 lb. of wet steam equals qh_{fg}, and the total heat in 1 lb. of wet steam equals $h_f + \frac{qh_{fg}}{100}$.

19. Steam Tables.—Table 4 shows the properties of dry saturated steam. More complete tables will be found in "Thermodynamic Properties of Steam," by J. H. Keenan and F. G. Keyes, and in the engineering handbooks. Column 1 gives the absolute pressure of the steam in pounds per square inch. Absolute pressure is the pressure shown on the steam gage plus the atmospheric or barometric pressure. (For sea-level barometer the atmospheric pressure is 14.696 lb. per sq. in.) Column 2 gives the corresponding temperature of the steam in degrees Fahrenheit. Columns 3 and 4 give in cubic feet per pound the specific volumes, respectively, of the liquid v_f, and of the saturated vapor v_g. The increase in volume resulting from evaporation v_{fg} may be obtained by taking the difference between columns 4 and 3 of v_g minus v_f. Columns 5, 6, and 7 show, respectively, the total heat or enthalpy of the liquid h_f, of evaporation h_{fg} (the latent heat), and of the saturated vapor h_g.

TABLE 4.—PROPERTIES OF SATURATED STEAM[1]

1	2	3	4	5	6	7
Absolute pressure, lb. per sq. in.	Temp., °F.	Specific volume, cu. ft. per lb.		Total heat or enthalpy, B.t.u. per lb.		
		Sat. liquid	Sat. vapor	Sat. liquid	Evap.	Sat. vapor
p	t	v_f	v_g	h_f	h_{fg}	h_g
0.50″ Hg	58.80	0.01604	1256.4	26.86	1060.6	1087.5
0.75″ Hg	70.43	0.01606	856.1	38.47	1054.0	1092.5
1.00″ Hg	79.03	0.01608	652.3	47.05	1049.2	1096.3
1.5″ Hg	91.72	0.01611	444.9	59.71	1042.0	1101.7
2.0″ Hg	101.14	0.01614	339.2	69.10	1036.6	1105.7
2.5″ Hg	108.71	0.01616	274.9	76.65	1032.3	1108.9
3.0″ Hg	115.06	0.01618	231.6	82.99	1028.6	1111.6
1.0	101.74	0.01614	333.6	69.70	1036.3	1106.0
2.0	126.08	0.01623	173.73	93.99	1022.2	1116.2
3.0	141.48	0.01630	118.71	109.37	1013.2	1122.6
4.0	152.97	0.01636	90.63	120.86	1006.4	1127.3
5.0	162.24	0.01640	73.52	130.13	1001.0	1131.1
6.0	170.06	0.01645	61.98	137.96	996.2	1134.2
7.0	176.85	0.01649	53.64	144.76	992.1	1136.9
8.0	182.86	0.01653	47.34	150.79	988.5	1139.3
9.0	188.28	0.01656	42.40	156.22	985.2	1141.4
10	193.21	0.01659	38.42	161.17	982.1	1143.3
11	197.75	0.01662	35.14	165.73	979.3	1145.0
12	201.96	0.01665	32.40	169.96	976.6	1146.6
13	205.88	0.01667	30.06	173.91	974.2	1148.1
14	209.56	0.01670	28.04	177.61	971.9	1149.5
14.696	212.00	0.01672	26.80	180.07	970.3	1150.4
16	216.32	0.01674	24.75	184.42	967.6	1152.0
18	222.41	0.01679	22.17	190.56	963.6	1154.2
20	227.96	0.01683	20.089	196.16	960.1	1156.3
22	233.07	0.01687	18.375	201.33	956.8	1158.1
24	237.82	0.01691	16.938	206.14	953.7	1159.8
26	242.25	0.01694	15.715	210.62	950.7	1161.3
28	246.41	0.01698	14.663	214.83	947.9	1162.7
30	250.33	0.01701	13.746	218.82	945.3	1164.1
40	267.25	0.01715	10.498	236.03	933.7	1169.7
50	281.01	0.01727	8.515	250.09	924.0	1174.1

TABLE 4.—PROPERTIES OF SATURATED STEAM.[1]—(*Concluded*)

1	2	3	4	5	6	7
Absolute pressure, lb. per sq. in.	Temp., °F.	Specific volume, cu. ft. per lb.		Total heat or enthalpy, B.t.u. per lb.		
		Sat. liquid	Sat. vapor	Sat. liquid	Evap.	Sat. vapor
p	t	v_f	v_g	h_f	h_{fg}	h_g
60	292.71	0.01738	7.175	262.09	915.5	1177.6
70	302.92	0.01748	6.206	272.61	907.9	1180.6
80	312.03	0.01757	5.472	282.02	901.1	1183.1
90	320.27	0.01766	4.896	290.56	894.7	1185.3
100	327.81	0.01774	4.432	298.40	888.8	1187.2
110	334.77	0.01782	4.049	305.66	883.2	1188.9
120	341.25	0.01789	3.728	312.44	877.9	1190.4
130	347.32	0.01796	3.455	318.81	872.9	1191.7
140	353.02	0.01802	3.220	324.82	868.2	1193.0
150	358.42	0.01809	3.015	330.51	863.6	1194.1
160	363.53	0.01815	2.834	335.93	859.2	1195.1
170	368.41	0.01822	2.675	341.09	854.9	1196.0
180	373.06	0.01827	2.532	346.03	850.8	1196.9
190	377.51	0.01833	2.404	350.79	846.8	1197.6
200	381.79	0.01839	2.288	355.36	843.0	1198.4
210	385.90	0.01844	2.183	359.77	839.2	1199.0
220	389.86	0.01850	2.087	364.02	835.6	1199.6
230	393.68	0.01854	1.9992	368.13	832.0	1200.1
240	397.37	0.01860	1.9183	372.12	828.5	1200.6
250	400.95	0.01865	1.8438	376.00	825.1	1201.1

[1] Reprinted, by permission, from "Thermodynamic Properties of Steam," by J. H. Keenan and F. G. Keyes, published by John Wiley & Sons, Inc., 1936.

AIR

20. Properties of Air.—In this chapter the properties of air are discussed briefly insofar as they are involved in the design of heating systems. In Chap. XIII the subject is dealt with more thoroughly in preparation for the study of air conditioning.

Perfectly *dry* air is a mixture of about 78 per cent of nitrogen by volume and 21 per cent of oxygen, with small amounts of several other gases. The ordinary atmosphere contains varying amounts of water vapor, in a superheated state, which is intimately mixed with the air just as the nitrogen, oxygen, and other gases are mixed with each other.

The properties of perfectly dry air are given in Table 5. The values for air containing moisture differ somewhat, but this table is sufficiently accurate for the calculations involved in heating work.

TABLE 5.—PROPERTIES OF DRY AIR[1]
(Barometric pressure 29.921 in.)

Temperature, °F.	Weight per cu. ft., lb.	Ratio to volume at 70°F.	B.t.u. absorbed by 1 cu. ft. dry air per °F.	Cu. ft. dry air warmed 1°F. per B.t.u.
60	0.07636	0.9811	0.01832	54.44
70	0.07492	1.0000	0.01798	55.62
80	0.07353	1.0189	0.01765	56.66
90	0.07219	1.0378	0.01733	57.70
100	0.07090	1.0567	0.01702	58.75
110	0.06966	1.0755	0.01672	59.81
120	0.06845	1.0945	0.01643	60.86
130	0.06729	1.1133	0.01616	61.88
140	0.06617	1.1322	0.01589	62.93
150	0.06509	1.1510	0.01563	63.98
160	0.06403	1.1701	0.01538	65.02
180	0.06203	1.2078	0.01490	67.11

[1] A.S.H. & V.E. "Guide," 1940; compiled by W. H. Severns. More complete tables are given in Chap. XIII.

21. Humidity.—The moisture content of the air is commonly called humidity. The *maximum* amount of humidity that can be present in air varies with the temperature. Air at 20° contains about 1.2 gr. of moisture per cu. ft. when fully saturated, whereas at 70° it contains 8 gr. when saturated (Fig. 4). In the condition of saturation the air contains the maximum possible amount of moisture existing as a vapor. (Saturation does not mean that condensed moisture, such as a fog, is present.)

Air when only partially saturated tends to absorb moisture from substances in contact with it and thus exerts a drying effect that is the more pronounced the less the moisture content of the air in relation to its saturation content at that temperature. The ratio of the actual moisture content of air to its saturation content at that temperature is called the **relative humidity.** When air is heated its relative humidity decreases, because the saturation content at the higher temperature is greater while the actual quantity of moisture remains the same. For example, 1 cu. ft. of air at 20° and containing approximately 1.08 gr. of moisture would have a relative humidity of 90 per cent. If heated to 70° it would have a relative humidity of but 12.2 per cent; and although its actual moisture content, or **absolute humidity,** would be unchanged, its drying effect would be greatly increased.

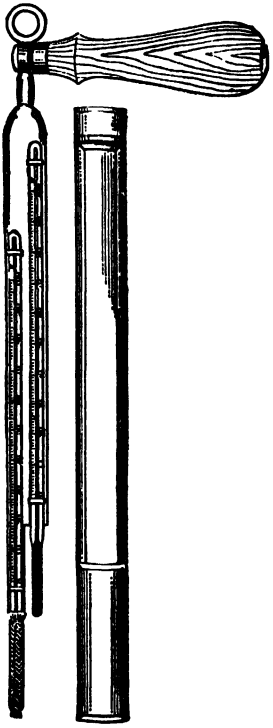

FIG. 3.—Sling psychrometer.[1]

22. Measurement of Humidity.—The relative humidity of air is readily found with the aid of the wet- and dry-bulb thermometers. The wet-bulb thermometer is an ordinary mercury-filled thermometer, the bulb of which is covered with cotton wicking. The end of the wicking extends into a bottle of water, and the entire wick is kept wet by absorption. When placed in air that is only partly saturated, water is continually evaporated, with an accompanying decrease in temperature of the wet-bulb thermometer as compared with the dry-bulb thermometer, the amount of the depression being definitely related to the moisture content. For accurate results there should be a rapid motion of the air over the wet bulb. For this reason the best form of wet- and dry-bulb thermometer for ordinary use is the sling psychrometer, illustrated in Fig. 3. In this instrument the wet- and dry-bulb thermometers are mounted on a metal strip pivoted to a handle. In using the instrument the wick surrounding the wet bulb is moistened, and the instrument is whirled rapidly and read at intervals until there is no further drop in the wet-bulb temperature.

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[1] From "Fan Engineering," Buffalo Forge Company.

It is necessary that the water used to moisten the wet bulb of the sling psychrometer be at approximately the wet-bulb temperature; otherwise the time required to bring the water to the wet-bulb temperature might be so great that parts of the wicking would become dry.

The ideal psychrometric chart in Fig. 4 is constructed for use with the sling psychrometer. This chart gives the moisture content of air in grains per cubic foot, the volume basis being the more convenient for ordinary ventilating work. In Chap. XIII is given a psychrometric chart that gives the properties of air on the basis of 1 lb. of air.

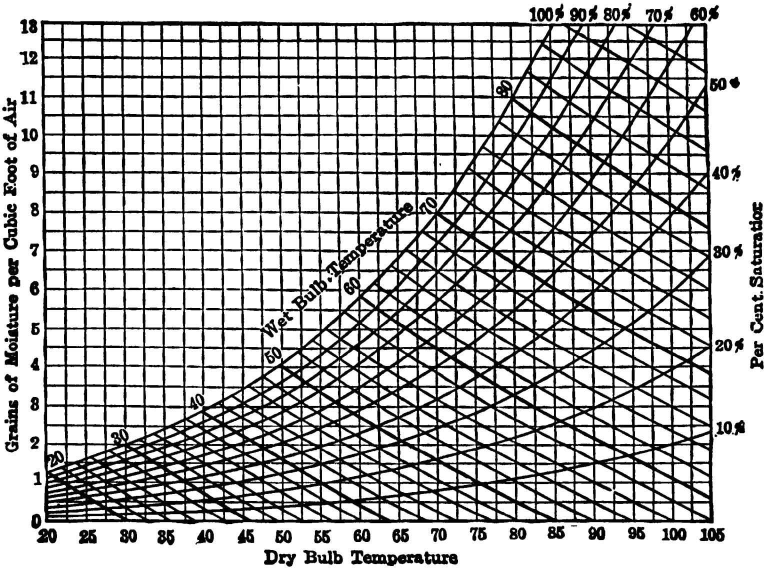

FIG. 4.—Psychrometric chart.[1]

The **dew point** of air having any given moisture content is the temperature at which the air would reach a state of saturation when cooled. When air is cooled to its dew point, the moisture begins to condense and to deposit on adjacent surfaces.

23. Example of Use of Psychrometric Chart.—Given a dry-bulb temperature of 80° and a wet-bulb temperature of 70°, find the relative and absolute humidity and the dew point. From the 80° point on the horizontal scale, follow the vertical line to its intersection with the diagonal line representing the wet-bulb temperature of 70°. In passing horizontally to the left from this point to the left-hand scale it will be noted that the absolute humidity is 6.65 gr. per cu. ft. To find the relative humidity, note that this same point lies between the 60 and 70 per cent relative-humidity lines (the curved lines extending upward to the right) and that the relative humidity is 62 per cent. To find the dew point, follow left horizontally from this same point to the curved line of wet-bulb temperatures, called the saturation line. The dew point is 64.5°.

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[1] For exact work the psychometric chart in Chap. XIII giving the moisture content in grains per *pound* of air should be used.

Bibliography

"Heat Transmission," by W. H. McAdams.
"Heat Engineering," by A. M. Greene, Jr.
"Introduction to Heat Transfer," by A. I. Brown and S. M. Marco.
"Thermodynamic Properties of Steam," by J. H. Keenan and F. G. Keyes.

Problems

1. Convert 50°F. to degrees centigrade. Convert 150°C. to degrees Fahrenheit. Convert 219°F. to degrees centigrade. Convert 225°F. to absolute temperature on the Fahrenheit scale.

2. A copper ball weighing 10 lb. is heated in a fire and immediately placed in a vessel of water having an equivalent water weight of 10 lb. The water is raised in temperature from 50 to 100°. What was the temperature of the ball when it was removed from the fire?

3. A bar of cast iron weighing 5 lb. and at a temperature of 250°F. and a bar of lead weighing 10 lb. and at a temperature of 300° are put into a tub of water that is at 120°. The water is heated to 123°. Neglecting the effect of the tub itself and the heat lost during the process, how much water is in the tub?

4. A piece of limestone weighing 10 lb. and at a temperature of 150°F and a piece of wrought iron weighing 20 lb. and at a temperature of 70° are put into a tank, and a sufficient quantity of water at 88° is added to bring the temperature of the water, stone, and iron to 90°. How much water is required, neglecting the heat lost during the process?

5. Compute the amount of heat conducted through a board 6 in. wide by 4 ft. long by 2 in. thick having a temperature on one side of 100° and on the other of 40°. Assume that $k = 1.0$.

6. Compute the amount of heat conducted per foot of length through a layer of asbestos insulation 2 in. thick applied over a pipe of 12 in. outside diameter. Inside surface of insulation is at a temperature of 220°; outside surface, at 130°. Assume that $k = 0.29$.

7. If 3520 B.t.u. is added to 3 lb. of water that is originally at a temperature of 120° and the pressure is maintained at 20 lb. per sq. in. abs., what are the resulting temperature and state of the fluid?

8. If 5 lb. (weight) of steam at 212° and 20 lb. of water at 60° are mixed at atmospheric pressure, what are the resulting temperature and state of the mixture?

9. If 10 lb. of steam at a pressure of 26 lb. per sq. in. abs. and a temperature of 290° is mixed with 10 lb. of water at 180°, the pressure being maintained constant, what is the resulting state of the mixture?

10. If 1,500 lb. of steam per hour at 16 lb. per sq. in. abs. are flowing through a pipe from which the heat loss is 46,750 B.t.u. per hr., what is the quality of the steam at the end of its travel?

11. How much heat must be added to 185 lb. of water at 42° to produce 185 lb. of steam at 10 lb. gage pressure and 98 per cent quality?

12. If steam at 160 lb. abs. and 97 per cent quality is reduced to 18 lb. abs. without gain or loss of heat, what is its state at the lower pressure?

13. If steam at 22 lb. abs. and 98 per cent quality is reduced to atmospheric pressure without gain or loss of heat, what is its state at the lower pressure?

14. Dry-bulb temperature 80°, wet-bulb 55°; find the relative humidity, absolute humidity, and dew point.

15. Wet bulb 60°, dew point 50°; what are the dry-bulb temperature, relative humidity, and absolute humidity?

16. If air has a dry-bulb temperature of 74° and a relative humidity of 62 per cent, how many degrees must it be cooled to reach its dew point?

17. If air has a dew point of 52°, how much moisture must be removed to reduce its dew point to 40°?

18. Given air at 30° and 40 per cent relative humidity, how much moisture must be added to give it a dew point of 54°?

19. Air at 40° dry bulb and 36° wet bulb is mixed with twice its volume of air at 80° and 68° dew point. What are the resulting dry-bulb and wet-bulb temperatures?

CHAPTER II

HEAT LOSSES FROM BUILDINGS

24. Sources of Heat Loss.—When the interior of a building is maintained at a temperature higher than that of the outside air, there is a continual loss of heat from the building, and it is the function of the heating system to supply sufficient heat to maintain the desired temperature. In a building that is heated intermittently, the heating system must also be capable of raising the temperature to the temperature desired for occupancy, in a reasonable time. Most heating systems, however, are actually designed for conditions of steady heat flow, and the requirements for intermittent heating are taken care of by arbitrarily adding capacity in unusual cases.

The calculation of the heat loss from each room in the building is of basic importance in heating-system design because it is used to determine the radiator sizes, which in turn fix the pipe sizes and the size of the boiler.

The loss of heat from a building takes place in several ways. Usually the greatest loss is through the walls and window glass. The heat flows through these materials at varying rates depending upon their conductivity and thickness, and is dissipated from the outer surfaces of the building by convection and by radiation. There is also a large amount of heat lost by the infiltration of air. From the surfaces buried in the ground—the basement walls and floors—heat is lost by conduction to the earth. There are also some minor sources of loss. The calculation of each of these items is made separately, and the aggregate loss for each room is thus obtained.

There is a wide variation in the degree of accuracy of the several parts of the heat-loss computation. The most exact is probably the calculation of heat flow through the walls, which is based on laboratory tests. The calculation of infiltration loss and the allowances for wind velocity and for exposure are only rough approximations, supported by limited test data. Therefore it should not be assumed that the computation of the heat loss from a building is highly exact, although the method hereinafter outlined gives results that are reasonably accurate and on the safe side. The computation in any given case should be made as accurately as possible and with due consideration of all predictable conditions.

25. Transmission through Walls.—In the simplest form of building, the walls consist of a solid section of a single material. The transmission of heat takes place from the air of the room. to the inner surface of the wall by convection, through the wall by conduction, and from the outer surface of the wall by convection and radiation. In order for heat to flow, there must be a difference of temperature from the inside to the outside. This temperature fall or gradient is represented by the line t—t_0, in Fig. 5. The temperature t_1 of the inside surface of the wall is lower than the air temperature t, and the temperature t_2 of the outside surface of the wall is higher than the outside air temperature t_0.

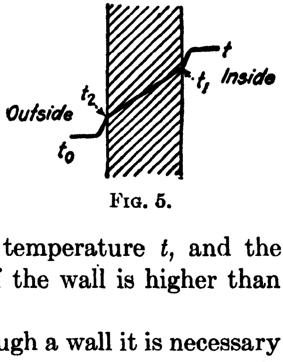

FIG. 5.

In order to compute the heat loss through a wall it is necessary to derive a method of expressing it in terms of t and t_0, the inside and outside temperatures. The heat loss can readily be expressed in terms of t_1 and t_2, the surface temperatures, as shown in Art. 9; but these temperatures are seldom known, although in an existing building they may be measured.

The flow of heat through the wall, as in Art. 9, is

$$H_c = A\frac{k}{x}(t_1 - t_2) \tag{1}$$

or, for an area of 1 sq. ft.

$$H_c = \frac{k}{x}(t_1 - t_2) \tag{2}$$

k being the conductivity of the material and x the wall thickness. The amount of heat passing from the inside air to the wall per hour per square foot of wall area is

$$H_1 = f_i(t - t_1) \tag{3}$$

where f_i is a constant that represents the **surface conductance,** that is, heat flow per square foot per degree difference in temperature between the air and the wall. Similarly, the heat leaving the outer surface of the wall per square foot of surface per hour is

$$H_2 = f_o(t_2 - t_o) \tag{4}$$

Now for a condition of steady flow the amount of heat flowing per hour through successive points of the wall must be the same. The flow through the wall expressed in terms of t and t_o is

$$H = U(t - t_o) \tag{5}$$

per square foot of wall area.

Since

$$H = H_c = H_1 = H_2$$

then

$$U(t - t_o) = f_i(t - t_1) = \frac{k}{x}(t_1 - t_2) = f_o(t_2 - t_o)$$

The quantity U is the over-all **transmission coefficient** for the wall—not to be confused with the internal conductivity of the material.

$$U(t - t_o) = f_i(t - t_1)$$
$$U(t - t_o) = \frac{k}{x}(t_1 - t_2)$$
$$U(t - t_o) = f_o(t_2 - t_o)$$

Transposing,

$$\frac{U}{f_i}(t - t_o) = t - t_1$$
$$\frac{U}{k/x}(t - t_o) = t_1 - t_2$$
$$\frac{U}{f_o}(t - t_o) = t_2 - t_o$$

Adding and simplifying,

$$U = \frac{1}{\frac{1}{f_i} + \frac{1}{f_o} + \frac{x}{k}} \tag{6}$$

The transmission coefficient U thus represents the over-all thermal transmittance of the wall including the surface resistances. The quantities f_i, f_o, and k/x are the conductivities of the respective elements of the wall, and their reciprocals may be considered as the *resistances* to heat flow.

The transmission coefficient of any kind of wall can thus be computed if the surface resistances and the conductivities of the various materials that make up the wall are known. In a wall consisting of two materials of conductivities k_1 and k_2, as shown in Fig. 6,

$$U = \frac{1}{\frac{1}{f_i} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{1}{f_o}} \tag{7}$$

For the wall construction shown in Fig. 7 it is necessary to add the surface conductances on either side of the air space. These are considered to be the same as f_i, and therefore

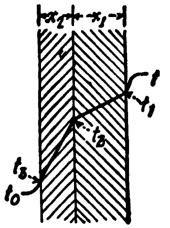

FIG. 6.

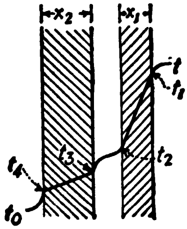

FIG. 7.

$$U = \frac{1}{\frac{1}{f_i} + \frac{x_1}{k_1} + \frac{1}{f_i} + \frac{1}{f_i} + \frac{x_2}{k_2} + \frac{1}{f_o}} = \frac{1}{\frac{3}{f_i} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{1}{f_o}} \tag{8}$$

The value of k, the conductivity, has been determined experimentally for a large number of materials.[1] A few of these are given in Table 6, and a more complete table is given in the Appendix. In some cases it is convenient to express the conductivity as the total conductance for the stated thickness. This is true of such materials as hollow tile, shingles, etc. The total conductance $C = k/x$, and in Eq. (6) the value $1/C$ would in such cases be used instead of x/k.

26. Surface Conductances.—The surface conductance f is affected by the nature of the surface and by the velocity and direction of the air movement over the surface. In general the rougher the surface the higher the conductance. Glass, for example, at a wind velocity of 20 m.p.h. has little more than half the surface conductance of stucco. The wind velocity has a marked effect, the conductance increasing with the velocity; but the wind direction has much less effect, and it is customary to use values of f for wind *parallel to* the surface, which is on the safe side, that condition giving somewhat higher values.

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[1] Tentative Method of Test for Thermal Conductivity of Materials by Means of the Guarded Hot Plate, A.S.T.M. Designation C 177-42 T.

TABLE 6.—INTERNAL CONDUCTIVITIES OF BUILDING MATERIALS[1]

Material	Conductivity[2] k	Total conductance C
Asbestos shingles	....	6.00
Brickwork	5.0	
Concrete, stone, 1-2 4 mix	12.0	
Hollow clay tile, 6 in.	....	0.64
Plaster, gypsum	3.3	
Wood lath and plaster	....	2.50
Woods:		
Maple or oak	1.15	
Yellow pine or fir	0.80	
Insulating materials:		
Cabots quilt	0.25	
Celotex	0.33	
Corkboard	0.30	
Rock wool	0.27	
Wood fiber	0.33	
Glass wool	0.27	
Aluminum foil as facing for both sides of air space over ¾ in. wide	....	0.46

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[1] A more complete table will be found in the Appendix.

[2] B.t.u. per hour per square foot of area per inch of thickness per degree difference in temperature.

Figure 8 shows the values of f as determined by tests. Values for f_i are taken as those for zero wind velocity, and values for f_o are taken for the wind velocity upon which the design is based, which is usually 15 m.p.h. Air spaces are considered to have zero wind velocity.

A concrete example of the effect of wind velocity on the temperature gradient through a wall is shown in Fig. 9. The existence of a wind, other conditions being constant, caused a marked decrease in the wall temperatures, both inside and outside.

27. Air Spaces.—Many types of wall construction contain air spaces, either for structural reasons or for the purpose of reducing the heat loss. Heat is transmitted across an air space by convection, by conduction, and by radiation. There is always a certain amount of air movement even in a so-called "dead" air space, and the convection currents pick up heat from the higher temperature surface and convey it to the lower temperature surface. The effect of convection decreases as the size of the air space is reduced and is greatest in vertical air spaces.

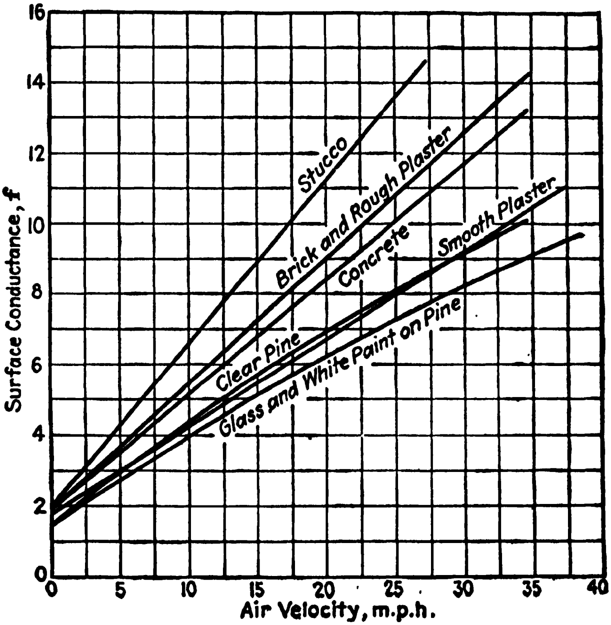

FIG. 8.—Surface conductances for various kinds of surfaces, for mean temperature of air and wall of 20°.[1]

The amount of heat transmitted by pure conduction across an air space is inversely proportional to the width of the space and becomes negligible in wide air spaces. The amount of heat transmitted by radiation depends upon the character of the surfaces and their relative temperatures. The process of heat transfer across an air space is thus very complex and not amenable to mathematical treatment. Rowley and Algren have obtained, in the laboratory, the results shown in Fig. 10 for vertical air spaces of 0.1 to 1.5 in. in width. The shape of these curves should be noted. They are nearly flat for thicknesses over 0.8 in. and theoretically approach infinity at zero thickness. The insulating value of an air space, though appreciable, is not so great as is often assumed.

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[1] From ROWLEY, ALGREN, and BLACKSHAW, "Surface Conductances as Affected by Air Velocity, Temperature, and Character of Surface," *Trans. A.S.H. & V.E.*, 1930.

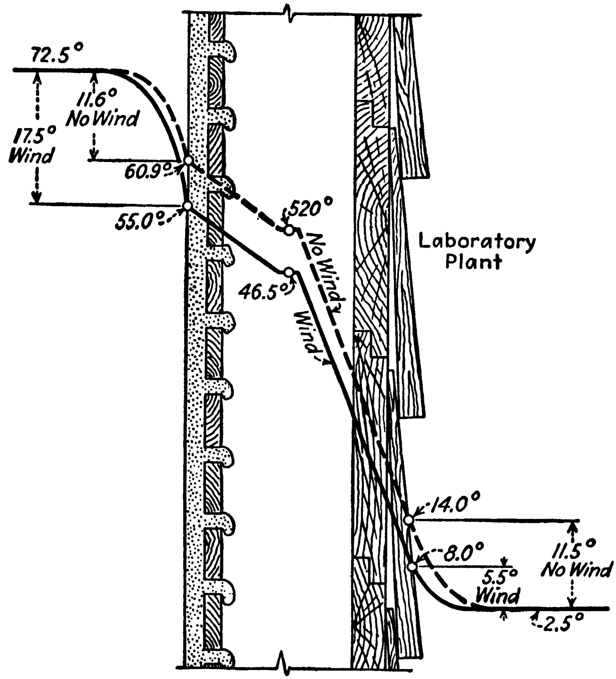

FIG. 9.—Temperature gradients through a wall (laboratory measurements).[1]

For air spaces over 1.5 in. in width the proper method of calculation is to use the surface conductances for the surfaces on the two sides of the air space as in Eq. (8).

28. Transmission Coefficients.—With the foregoing data on the values of k and f it is possible to calculate with reasonable accuracy the value of U, the over-all transmission coefficient, for any wall construction. Although most of the ordinary types of construction have been tested for heat transmission, unusual forms are frequently encountered.

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[1] From WILLARD and KRATZ, "Wall Surface Temperatures," *Trans. A.S.H. & V.E.*, 1930.

It is a simple matter to write an equation similar to Eq. (6), (7), or (8) for the wall in question and to insert therein the proper conductivities and surface conductances.

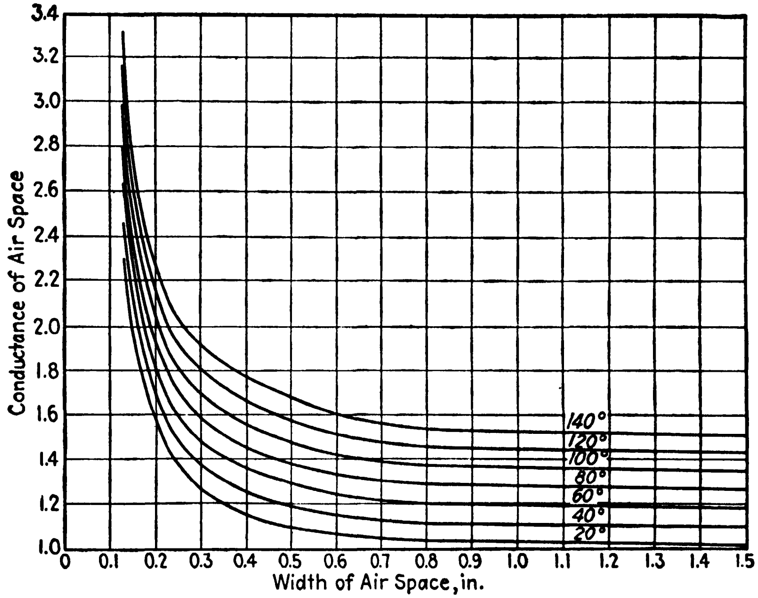

FIG. 10.—Conductance of air spaces for various mean temperatures.[1]

Example 1.—Assume a wall construction consisting of 13 in. of brickwork and 1 in. of plaster (on inside surface). For brickwork, $k = 5.0$; for plaster, $k = 3.3$; $f_i = 1.9$; f_o for 15-m.p.h. wind $= 7.1$ (Fig. 8).

$$U = \frac{1}{\frac{1}{1.9} + \frac{1}{3.3} + \frac{13}{5.0} + \frac{1}{7.1}} = 0.280$$

Example 2.—Assume a wall construction as shown in Fig. 11, consisting of plaster, corkboard, air space (formed by the studding), sheathing, and brick.

$$U_1 = \frac{1}{\frac{1}{1.9} + \frac{0.5}{3.3} + \frac{1}{0.30} + \frac{1}{1.9} + \frac{1}{1.9} + \frac{1}{0.80} + \frac{4}{5.00} + \frac{1}{7.1}}$$

$U_1 = 0.139$ for the space between the studding

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[1] From ROWLEY and ALGREN, "Thermal Resistances of Air Spaces," *Trans. A.S.H. & V.E.*, 1929.

For the area covered by the studding (which is $3^{11}/_{16}$ in. thick)

$$U_2 = \frac{1}{\frac{1}{1.9} + \frac{0.5}{3.3} + \frac{1}{0.30} + \frac{3.69}{0.80} + \frac{1}{0.80} + \frac{4}{5.00} + \frac{1}{7.1}} = 0.092$$

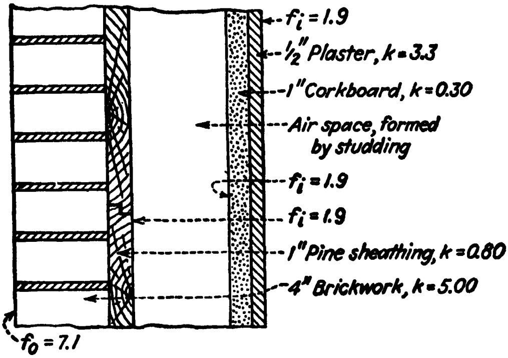

Fig. 11.

The studding is on 16-in. centers, and the width of the studding is $1^{11}/_{16}$ in. Therefore

$$U = \frac{U_1 \times 14^5/_{16} + U_2 \times 1^{11}/_{16}}{16}$$

$$U = \frac{0.139 \times 14.31 + 0.092 \times 1.69}{16} = 0.134$$

29. Experimental Determination of Transmission Coefficients. A great many of the common forms of building construction have been tested for heat transmission. The method most commonly used in making such determinations is to employ a cubical box with five of its six faces made of a material of low conductivity and guarded from heat loss by a second box surrounding it and maintained at nearly the same temperature. The sixth face is constructed of the material to be tested. The temperature inside the box is maintained constant and above that of the surrounding air, by supplying a measured amount of heat, usually electrically, to the interior. With the proper corrections made for the slight loss through the other five sides, the heat transfer through the material under test can be accurately determined.

In Table 7 are given the values of U for several common types of building construction. More complete tables will be found in the Appendix. These values are largely calculated values, but in several cases they have been confirmed by laboratory tests.

30. Insulation.—The effect of insulation on the heat loss through a wall can be readily appreciated by noting the magnitude of the third term in the denominator in Example 2, Art. 28, which indicates the resistance of the insulation. Unless the heating season is short or a very cheap fuel is used, properly installed insulation is usually financially justified. It also has some effect in making a building more comfortable through creating a more even temperature, by reducing heat flow and by increasing the interior surface temperatures so as to reduce the radiation heat loss from the human body. In summer it keeps a building cooler during short periods of high temperatures but obviously cannot be effective for long periods.

TABLE 7.—COEFFICIENTS OF HEAT TRANSMISSION FOR VARIOUS FORMS OF BUILDING CONSTRUCTION

Building Construction	U, B.t.u. per Sq. Ft. per Hr, per Deg. Difference in Temperature
Walls:	
Brick wall, 8 in. thick, plain	0.50
Brick wall, 12 in. thick, plain	0.35
Brick wall, 8 in. thick, ½ in. plaster on wall	0.46
Brick wall, 12 in. thick, ½ in. plaster on wall	0.34
Hollow tile, 8 in. thick, stucco, ½ in. plaster	0.37
Clapboard, sheathing, studs, wood lath, and plaster	0.25
Clapboard, with ½ in. rigid insulation in place of lath	0.19
Ceilings and roofs:	
Wood lath and plaster, no floor above	0.62
Wood lath and plaster, joists, 1 in. pine floor above	0.29
Wood shingles, on wood strips, rafters, no ceiling	0.48
Concrete roof, 4 in., metal lath and plaster	0.40
Windows and doors:	
Ordinary windows	1.13
Double windows	0.45
Wood door, 1½ in. nominal thickness	0.52

There are several types of insulating materials, which are best described as follows:

1. Materials used as the base for plastering (such as the corkboard in Fig. 11).
2. Loose-fill materials poured into the studding space and similar spaces.
3. Batts, or blankets, which are soft materials supplied as flexible sheets and fastened between the studding.

4. Sheathing materials, used in place of the ordinary shiplap outside the studding or in similar locations.

5. Reflective-type insulation. This is usually metallic foil and depends for its effect not on low conductivity but on its high reflectivity and low emissivity. It is installed in the studding space and sometimes arranged so as to divide the original air space into two or more vertical layers. In addition to its effect it thus adds two surface resistances for each sheet of foil and thereby increases the insulating value.

The insulating value of most insulating materials has been determined in the laboratory on laboratory samples, but there is much discussion as to how greatly they are affected by conditions of actual use such as moisture, settlement, and (in the case of reflective foils) corrosion and dust.

31. Condensation in Building Structures.—The present trend toward the maintaining of artificial humidities is creating an important problem in building construction. Where high humidities (35 to 40 per cent) are maintained, the moisture tends to travel through the building structure to the outdoors or to cooler spaces in the building.

As air travels through air spaces formed by the building structure and reaches a region where the temperature is below the dew point of the air, the moisture condenses, and appreciable quantities of moisture or frost may accumulate. This type of condensation is frequently referred to as interstitial or structural condensation because of its location. Any moisture condensing within an insulation reduces the insulating effect of the material and may cause damage to the wall, such as blistering and peeling of exterior paint or staining of the wallpaper. Structural condensation may also appear in attic spaces beneath the roof boards and sometimes below the flooring where attic flooring is used.

With no moisture barrier provided in the wall, the use of insulation may actually aggravate the trouble. For example, if insulation that is not impervious to moisture is used as a plaster base with ordinary frame-wall construction, it naturally lowers the temperature in the studding space. Air at 70° and 35 per cent humidity has a dew point of 39°. If there were little infiltration of air from out-of-doors into the studding space, the air in that space would have nearly the same dew point. The temperature in the studding space in zero weather would undoubtedly be less than 39° (probably between 20 and 30°), and condensation would take place. Actually the possibilities of condensation are much greater for an insulated wall than for a noninsulated wall unless some type of vapor barrier is used on the *warm* side of the wall.

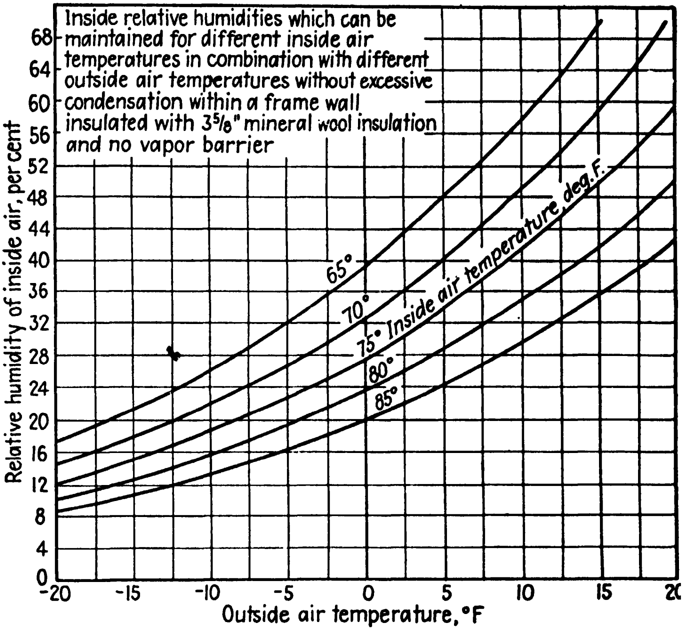

FIG. 12.—Limiting relative humidities which can be maintained without excessive wall condensation.

Figure 12[1] shows the maximum inside relative humidity that may be maintained with different inside-air temperatures without danger of excessive condensation within a frame wall insulated with 3⅝ in. of mineral wool and with the vapor barrier omitted. For −10° outdoor temperature and 70° inside-air temperature, the maximum relative humidity that may be maintained without excess in structural condensation is 22 per cent. If the outdoor temperature is 5° and the inside temperature is

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[1] ROWLEY, JORDAN, and LUND, "Conservation of Fuel," *Univ. Minn. Eng. Exp. Sta. Bull.* 20, 1943.

70°, the interior relative humidity may be maintained at 40 per cent without any serious consequences.

With new construction, the simplest method of preventing condensation is to apply a vapor barrier consisting of a paper with a continuous coating of asphalt on the *warm* side of all outside exposed walls and ceilings. Many of the batt or blanket insulations now have such a vapor barrier attached.

In old construction where no vapor barriers have been installed, it is possible to apply a paint having a high moisture-travel resistance to the inside surface of the plaster. A typical composition of such a paint consists of 40 per cent pigment and 60 per cent vehicle, with the pigment composition 100 per cent titanium calcium. A typical vehicle composition of such a paint is 16.7 per cent resin, 28.8 per cent vegetable oil, and 54.5 per cent drier and thinner. Any aluminum paint that produces a bright surface is also suitable for this purpose, provided a vehicle of water-resisting spar varnish with a minimum of 40 per cent non-volatiles is used. Some surface-treated canvas wall coverings serve as satisfactory vapor barriers. Ordinary wallpaper and plain plaster have little resistance to vapor travel.

Where no vapor barrier has been installed in the ceiling and condensation in the attic occurs, it is possible to reduce or entirely eliminate such difficulties by the addition of attic ventilation. The ventilation of attic spaces in structures where the top-floor ceiling has been insulated does not materially increase the heat loss of the building. The openings for attic ventilation should be located to secure cross ventilation and be so proportioned to allow 0.25 sq. in. per sq. ft. of ceiling area.[1]

The exact conditions under which a vapor barrier is actually necessary have not as yet been clearly determined. In general, however, it seems to be agreed that with insulated walls, in the Northern states of the United States (or colder locations), a vapor barrier should be provided.[2]

32. Condensation on Windows.—Condensation forms on the inside surface of windows when the air in contact with the glass becomes chilled below its dew point. Naturally this occurs more readily if the room air is artificially humidified and thus has a higher dew-point temperature. The increasing use of humidifying devices and the high humidities often carried in air-conditioned buildings bring this condition into prominence.

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[1] Rowley, Algren, and Lund, "Condensation of Moisture and its Relation to Building Construction and Operation," *Trans. A.S.H. & V.E.*, 1939.

[2] Teesdale, "How to Overcome Condensation in Building Walls and Attics," *Heating & Ventilating*, April, 1939; Wooley, "Moisture Condensation in Building Walls," Nat. Bur. Standards, BMS 52, July, 1940.

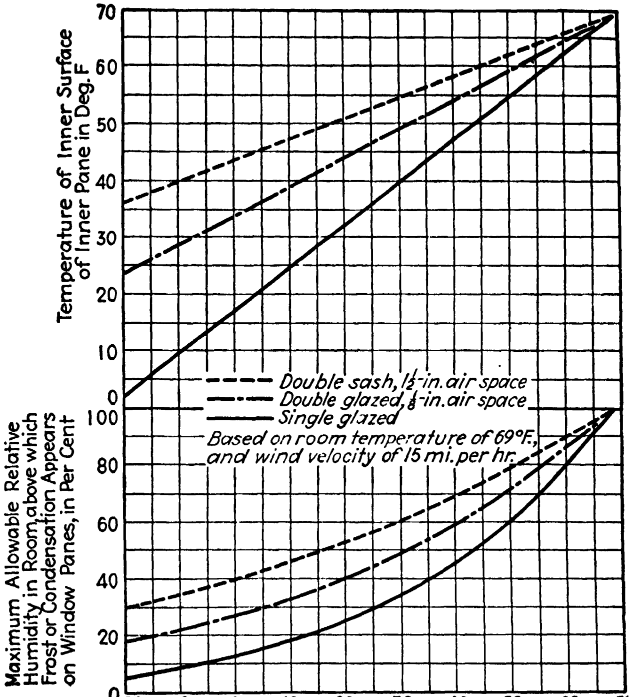

FIG. 13.—Formation of condensation on windows.[1]

Condensation on windows is unsightly and may damage woodwork and decorations. Double windows or double glazing are highly desirable if artificial humidification is to be used.

The space between the two windows should be vented to the *outside*. If this is not done, the water vapor will build up in that space, and condensation will form on the outer glass.[2] This can usually be accomplished in the case of tight-fitting storm sashes by drilling a small hole through the sash.

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[1] From "Gravity Warm Air Furnace Heating," edited by J. D. Hoffman.

[2] LEONHARD and GRANT, "Frost and Condensation on Windows," *Trans. A.S.H. & V.E.*, 1929.

Hermetically sealed windows with two or more panes of glass are now available. A metal-to-glass seal bonds the panes together to prevent dirt and moisture from entering the air space. The dry air sealed within the panes prevents condensation from forming on the inner surfaces of the glass. The curves in Fig. 13 show the probable glass-surface temperatures and the probable humidities at which condensation will form, for various outside temperatures. These are calculated results. The amount of inward leakage around the sash and the amount of air movement in the room have considerable effect. Both tend to reduce condensation.

33. The Nicholls Heat Meter.—An interesting apparatus for measuring the heat flow through the walls of actual buildings is the heat meter devised by Percy Nicholls.[1] It consists of a plate of some material such as cork or formica and having a number of thermocouples regularly arranged on each of its faces. It is carefully calibrated to obtain the relation of the heat flow to the temperature gradient through it, and when it is placed on the inside surface of a wall the heat flowing through the plate and consequently through the wall may be measured. Although there are certain inaccuracies in its use, it affords a simple means for investigating conditions in existing buildings.

34. Infiltration.—No building is perfectly airtight; there is a large amount of leakage through the cracks around windows and doors, through construction joints, and even directly through the walls in some cases. The amount of this infiltration depends largely upon how well the building has been constructed and upon the type of construction.

One method of computing the effect of infiltration is to assume that the volume of air in the room or building is renewed a certain number of times per hour. This method is simple and sufficiently

TABLE 8.—AIR CHANGES IN DIFFERENT KINDS OF ROOMS

	Changes per Hour
Rooms with no windows or outside doors	½
Rooms with windows on one or two sides	1 –1½
Rooms with windows on three or four sides	2
Entrance halls	2 –3
Small retail stores	2 –3
Factories, lofts, etc	2 –3

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[1] *Trans. A.S.H. & V.E.*, 1924. See also DUNKLE, "Heat Meters," *Bull Am. Meteor. Soc.*, March, 1940.

accurate in some cases. The number of air changes usually assumed are given in Table 8.

The second and preferable method is to compute the length of the crack around the windows and to assume an amount of air leakage per foot of crack depending on the window construction, based on tests of window leakage by various investigators (Table 9). Obviously it would not be correct to use the aggregate length of cracks of the windows on all sides of a room, because air is usually flowing inward through some of them but outward through others. The generally accepted method is to take, in a room having one exposed wall, the total crack; in a room having two exposed walls, the amount of crack in the wall having the greater amount; and with three or four exposed walls, the amount in the wall having the greatest amount, but in no case less than half of the total crack. This method is empirical and not very accurate, but the conditions are so varied that

TABLE 9.—INFILTRATION DUE TO WINDOWS OF VARIOUS TYPES[1]

Type of window	Infiltration, cu. ft. of air per hour per foot of crack		
	Wind velocity, 5 m.p.h.	Wind velocity, 15 m.p.h.	Wind velocity, 30 m.p.h.
Double-hung, wood-sash windows:			
Per foot of crack around sash and meeting rail of 1/16 in. width, approx. Window unlocked, without weather strips	6.6	39.3	103.7
With weather strips	4.3	23.6	63.4
Double-hung, metal-sash windows:			
Per foot of crack around sash and meeting rail. No weather strips, window locked	20	70	154
No weather strips, window unlocked	20	74	170
Weather strips, window unlocked	6	32	76
Rolled-section steel windows:			
Per foot of crack around ventilating sash...			
Industrial type, 1/16 in. crack	52	176	372
Architectural type, 3/64 in. crack	20	88	208
Residential type, 1/32 in. crack	14	52	128
Heavy casement type, 1/32 in. crack	8	38	96

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[1] These values, abstracted from the A.S.H. & V.E. "Guide," 1945, are based on tests by various investigators but are approximately 20 per cent less than the actual test results, to allow for the probable reduction in leakage in an actual building due to the building up of pressure within the building.

no exact method could be devised. The amount of infiltration thus computed should be checked against the figures as determined from Table 8. If the room has windows or outside doors, the infiltration finally assumed should not be less than the equivalent of one-half of an air change per hour..

Storm sashes are often not very tight-fitting and therefore may not greatly reduce the infiltration if the original window itself is well fitted or is weather-stripped. But if the window is poorly fitted and the storm sash is tight, the bottleneck, so to speak, is at the storm sash, and the infiltration is much reduced. It is impracticable to set up any definite standards, but tests indicate that for a loose window and a fastened storm sash the reduction of infiltration due to the storm sash may be 60 per cent; whereas with a fairly tight window, not weather-stripped, the same storm sash may reduce the leakage 20 per cent.

TABLE 10.—INFILTRATION THROUGH WALLS[1]

Type of wall	Infiltration, cu. ft. of air per square foot of wall per hour		
	Wind velocity, 5 m.p.h.	Wind velocity, 15 m.p.h.	Wind velocity, 30 m.p.h.
Brick wall, 8½ in. thick, plain	1.75	7.85	22.9
Brick wall, 8½ in. thick, plastered	0.017	0.066	0.236
Brick wall, 13 in. thick, plain	1.44	7.48	21.2
Frame wall with lath and plaster and building paper	0.02	0.12	0.31

[1] Larson, Nelson, and Braatz, *Trans A. S. H. & V. E.*, 1929, 1930. The values given are reasonable design figures, but the actual leakage will depend greatly upon the care with which the wall is constructed.

35. Infiltration through Walls.—The amount of air leakage through a wall which is plastered or which contains building paper, properly applied, is usually so small that it may be neglected; but for a rough brick wall, air leakage may be an important item. Table 10 gives the leakage that may be expected.

36. Heat Loss by Infiltration.—The heat required to supply the infiltration losses must be sufficient to warm the air from the temperature of the outside air to that of the room. If the infiltration is figured on the basis of a certain number of air changes per hour, the loss from this source may be expected as follows:

Let

H_s = heat required per hr. to supply loss due to infiltration.
C = cubic contents of the room.
n = number of changes per hr.
t = temperature of the room.
t_0 = temperature of the outside air.

$$H_s = \frac{C(t - t_o)n}{55.2}. \qquad (9)$$

The factor $55.2 = 1/(0.2415 \times 0.0749)$ = number of cubic feet of air that 1 B.t.u. will raise 1°, where 0.2415 is the specific heat of air at constant pressure and 0.0749 is the weight of 1 cu. ft. of dry air at 70° (Table 5, Chap. I).

If the infiltration is computed in cubic feet of air per hour, then

$$H_s = \frac{\text{cu. ft. per hr.} \times (t - t_o)}{55.2} \qquad (10)$$

The heat required for ventilation when a building is artificially ventilated can be computed in the same manner.

37. Moisture Filtration.—The heat equivalent for the infiltration loss calculated from Eqs. (9) and (10) is often referred to as the *sensible heat loss*, due to the fact that it involves only the dry-air heat loss. From the discussion in Art. 21 it was shown that unless moisture is added to the outdoor air leaking into an enclosure during the winter, a low relative humidity is likely to prevail. If equipment is provided for humidifying this air, it is necessary to compute the *latent heat loss* and to add this equivalent to the sensible losses in order to obtain the total infiltration heat loss, as follows:

Let

H_l = latent heat required per hr. to maintain desired indoor relative humidity.
Q = quantity outside air leaking into enclosure, cu. ft. per hr.
w = weight water vapor inside air, gr. per cu. ft.
w_o = weight water vapor outdoor air, gr. per cu. ft.
h_{fg} = latent heat of vapor at w, B.t.u. per lb.
7,000 = gr. per lb.

$$H_l = Q\left(\frac{w - w_o}{7{,}000}\right) h_{fg} \qquad (11)$$

Example 1.—Suppose a residence having 10,000 cu. ft. of occupied space is to be maintained at a dry-bulb temperature of 70° and a relative humidity of 40 per cent. If the outdoor temperature is 20° with 80 per cent relative humidity and the house has one air change per hour, compute the latent heat loss and the gallons of water required per day for humidification.

Substituting in Eq. (11), $w = 3.2$ gr. per cu. ft. (from Fig. 4), $w_o = 1.0$ gr. per cu. ft., and $h_{fg} = 1054.3$ (Table 65).

$$H_l = 10{,}000 \times 1 \left(\frac{3.2 - 1.0}{7{,}000}\right) 1054.3 = 3313.5 \text{ B.t.u. per hr.}$$

The water to be added per 24 hr. is calculated from the previous equation as follows: 3313.5 ÷ 1054.3 = 3.14 lb. water vapor required per hr. or 3.14 × 24 = 75.36 lb. per day. From Table 58, the density of water supplied is 62.3 lb. per cu. ft., and the quantity required per day is 75.36 ÷ 62.3 = 1.21 cu. ft. × 7.48 gal. per cu. ft. = 9.05 gal. This is equivalent to an hourly evaporation rate 9.05 ÷ 24 = 0.377 gal. or 0.377 × 4 = 1.508 quarts water per hr.

In ordinary practice the moisture filtration loss is not usually taken into account; but it is necessary to do so in some cases, particularly if there is large leakage and a high inside humidity. The chief value of the foregoing example is to illustrate the kind of calculation to be made, if necessary.

38. Infiltration in Tall Buildings.—There is a pronounced chimney effect in tall buildings, owing to the difference in density of the warm inside air and the cold air outside. This tends to increase the infiltration at the lower floors and to cause an outward flow of air at the upper floors, whereas near the mid-height there exists a neutral zone at which there is no tendency for air flow either in or out.[1] The chimney effect may be minimized by closing, as tightly as possible, all openings from floor to floor, such as stairways and elevator shafts. This is imperative in very tall buildings, or the lower floors would be almost uninhabitable. In spite of this precaution there is still need of supplying a greater amount of heat to the lower floors.

If the neutral zone be assumed to be at mid-height, it is a simple matter to calculate the static pressure at various elevations. It is more usual, however, to use the following empirical formulas as given in the A.S.H. & V.E. "Guide":

$$M_e = \sqrt{M^2 - 1.75a} \tag{12}$$

$$M_e = \sqrt{M^2 + 1.75b} \tag{13}$$

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[1] See J. E. Emswiler, "The Neutral Zone in Ventilation," *Trans. A.S.H. & V.E.*, 1926.

where M_e = equivalent wind velocity to be used with Tables 9 and 10.

M = wind velocity assumed for buildings of ordinary height.

a = distance of window above mid-height.

b = distance of window below mid-height.

These formulas are not adequately verified by actual test data, although one series of tests has indicated that they are probably fairly correct.[1]

39. Inside Temperatures.—The heat losses are computed on the basis of certain assumed inside and outside temperatures [t and t_o in Eq. (5), Art. 25].

TABLE 11.—TEMPERATURES OF UNHEATED SPACES

Value of U for partition or ceiling	Temperature of unheated space to be assumed, °F.			
	Design outside temperature, °F.			
	+10	0	−10	−20
0.6	41	35	30	25
0.5	38	32	26	21
0.4	34	28	22	15
0.3	30	23	16	9
0.2	25	17	10	2
0.1	18	10	1	− 8

In determining the heat loss from a room all surfaces should be considered that have on the outside a lower temperature than the temperature to be maintained in the room. If the room is over an unheated portion of the basement or below an unheated attic, the loss through the floor or ceiling should be considered. Similarly, if an adjacent room is liable to be unheated at times, the additional heat loss through the wall should be taken into account. Temperatures to be assumed for unheated spaces such as attics or unheated rooms are given in Table 11. Ordinarily it is assumed that there is no loss through inside walls where the surrounding rooms are heated.

The inside temperatures ordinarily assumed for different types of buildings at the breathing line, 5 ft. above the floor, are given in Table 12.

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[1] AXEL MARIN, "Influence of Stack Effect on Heat Loss in Tall Buildings," *Trans. A.S.H. & V.E.*, 1934.

However, the heat losses should not be based on these breathing-line temperatures in all cases. In a high-ceilinged room the temperature at the higher levels may be appreciably above the breathing-line temperatures, and the heat losses from those portions of the wall and glass areas may thus be correspondingly greater.[1] An arbitrary rule sometimes used in special cases is to allow 2 per cent increase in temperature per foot above the 5-ft. level, for rooms heated with ordinary radiators. Where the air circulation is increased, as by a fan system, the allowance is 1 per cent per ft. For rooms of ordinary height no such allowance is made.

TABLE 12.—INSIDE TEMPERATURES

Building	Rooms	Degrees
Churches	Auditorium	65–68
Factories	Foundries	50–60
	Machine shops	60–65
Garages	....	50–55
Greenhouses	....	78
Hospitals	Kitchens	66
	Operating rooms	70–95
	Private rooms	70–72
	Wards	68
Hotels	Ballrooms	65–68
	Bedrooms	70
	Dining rooms	70
Office buildings	....	68–72
Residences	Living quarters	70–72
	Bathrooms	80–85
Restaurants	....	65–70
Schools	Auditoriums	68–70
	Classrooms	70–72
	Gymnasiums	55–65
Stores	Drug	65–70
	Grocery	65
	Wearing apparel	72
Theaters	Auditorium	68–70
	Lounge	68–72

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[1] LARSON, NELSON, and JAMES, "Tests of Three Heating Systems in an Industrial Type of Building," *Trans. A.S.H. & V.E.*, 1935.

On the other hand, the temperature at the floor, which of course affects the heat loss through the floor, is lower than the breathing-line temperature, and a difference here of 5° is usually assumed.

40. Additional Allowances.—From years of none-too-reliable experience there has grown a practice of adding an allowance to the calculated heat loss on the sides of the building exposed to the prevailing winter winds. This exposure allowance, when used, is ordinarily 15 per cent. It is difficult to defend this practice, inasmuch as the transmission losses and the infiltration are already figured for a 15-mile, or higher, wind velocity. Exposure allowances are arbitrary safety factors, which were more necessary in former years than they are today when methods of calculation are more exact, and the present tendency is away from the practice. It is certain that there actually is an increased heat loss on the windward side of a building; but whether or not the radiator surface should be increased on that side, and if so to what extent, is still a debatable point.

Another correction factor sometimes required is to provide for warming up a building quickly after it has been allowed to cool down. The design of a heating system based only on the heat loss after the conditions have reached a steady state obviously does not provide for the considerable amount of heat required to warm the building structure itself and its contents. For ordinary conditions of intermittent heating, as when the heat supply is shut off for periods during the day or even for the entire night, there is usually enough margin of capacity to take care of this on all but the very coldest days. Only infrequently must an allowance be made, as in a building that may be entirely unheated for periods of several days in severe weather. In such cases it is desirable to add 10 or 15 per cent to the calculated heat loss.

41. Outside Temperatures and Wind Velocities.—The heat losses are affected both by the outside temperature and by the wind velocity, and their relative importance varies with the design of the building. If a building were, say, entirely of brick, with no windows, the effect of the wind would be merely to increase the exterior surface conductance, which is a minor factor, and the outside temperature would control the heat loss. On the other hand, a building with a large number of leaky windows would have a much larger heat loss with high wind velocity. Here the outside temperature would also be important for two reasons: It affects the heat loss through the walls and glass, and it affects the heat loss per cubic foot of infiltration.

Thus in ordinary buildings it would be desirable, since the design is based on the maximum heating load, to choose the worst *combination* of temperature and wind velocity that might be expected, but this is hardly practicable.

It is customary to select as the design temperature not the lowest temperature on record for the given locality but a figure at least 15° above the absolute minimum. The thermal capacity of the structure is usually sufficient to carry over the extreme lows, which ordinarily occur in the night hours. Local practice varies greatly in this respect and should be ascertained before a heating system is designed.

Customary design temperatures for a few selected localities are given in Table 13. Since design temperatures are generally specified in 5° intervals, it will be observed that these values check closely with the rule, provided that the design value be 15°F. above the lowest temperature. Close[1] found that these recommended design temperatures checked closely with the average of the minimum temperatures that have occurred once each winter for a period from 1899 to 1938.

As to the wind movement to be chosen, the common practice is to use, for calculating the infiltration loss, a local average for December, January, and February. For determining the surface conductance it is sufficiently accurate to assume a velocity of 15 m.p.h. This subject of wind movement needs, and is now receiving further study and observation under actual working conditions.

The moisture content, consistency, and heat capacity of soil greatly affect the ground temperature at various depths, but for heat-loss determinations it can generally be assumed that the ground temperature under a basement floor will average about 50° in most localities. The ground temperature surrounding a basement wall below grade may be estimated as the average temperature between 50° and the prevailing outdoor design temperature. Tests[1] have indicated that a coefficient of $U = 0.10$ may be used for all types of concrete floors on the ground and for basement walls below grade, with or without insulation. This coefficient is much lower than previously recommended values, but it can be tentatively justified on the basis that the ground temperatures recorded in this investigation showed that the surrounding earth offered more resistance to heat flow than was formerly estimated. Results also indicated that the heat penetrated to considerable depth in the earth and the effectiveness of the insulation was therefore minimized when the over-all resistances of the concrete, insulation, and earth were all considered together. However, it was found that the insulation may be effective in raising the inside surface temperature for the purpose of eliminating troublesome condensation.

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[1] P. D. Close, "Selecting Winter Design Temperatures," *Jour. A.S.H.&V.E.*, July, 1944.

TABLE 13.—WEATHER BUREAU RECORDS OF TEMPERATURES AND WIND

City	Average temperature, Oct. 1–May 1, °F.	Lowest temperature ever reported, °F.	Design temperature usually assumed, °F.	Average wind velocity Dec., Jan., Feb., m p.h.	Direction of prevailing wind, Dec., Jan. Feb.
Atlanta	51.5	− 8	+10	11 4	NW
Boston	38.1	−18	0	11.2	W
Chicago	36.4	−23	−10	12.5	W
Cleveland	37.2	−17	− 5	15.0	SW
Denver	38.9	−29	−15	7.5	S
Detroit	35.8	−24	−10	12.7	SW
Minneapolis	29.4	−34	−20	11.3	NW
New York	40.7	−14	0	16.7	NW
Philadelphia	42.7	−11	0	11.0	NW
San Francisco	54.2	27	+30	7.5	N
St. Louis	43.6	−22	− 5	11.7	S
Washington, D.C.	43.4	−15	0	7.9	NW

42. Intermittent Heating.—It is desirable next to consider what takes place when a building is heated intermittently. It is a fact, which can be demonstrated both theoretically and by actual test, that a great deal of heat can be conserved by cutting off the heat supply in any type of building during periods of nonoccupancy.

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[1] HOUGHTEN, TAIMUTY, GUTBERLET, and BROWN, "Heat Losses through Basement Walls and Floors," *Trans. A.S.H. & V.E.*, 1942.

Referring to Fig. 14, let the ordinate *AB* represent the rate of heat loss from a building during the heated period, the outdoor temperature being assumed to be constant. The rate of heat delivery by the heating system will also be equal to *AB*. At *C* the heat supply is entirely shut off, and the heat delivered by the heating system falls off along the line *CD* as the radiators and piping cool and soon reaches zero. The heat loss from the building falls off along the line *CE*, and the inside temperature drops as indicated by the upper curve in the figure. The area *CDEF* represents the heat given up by the building structure.

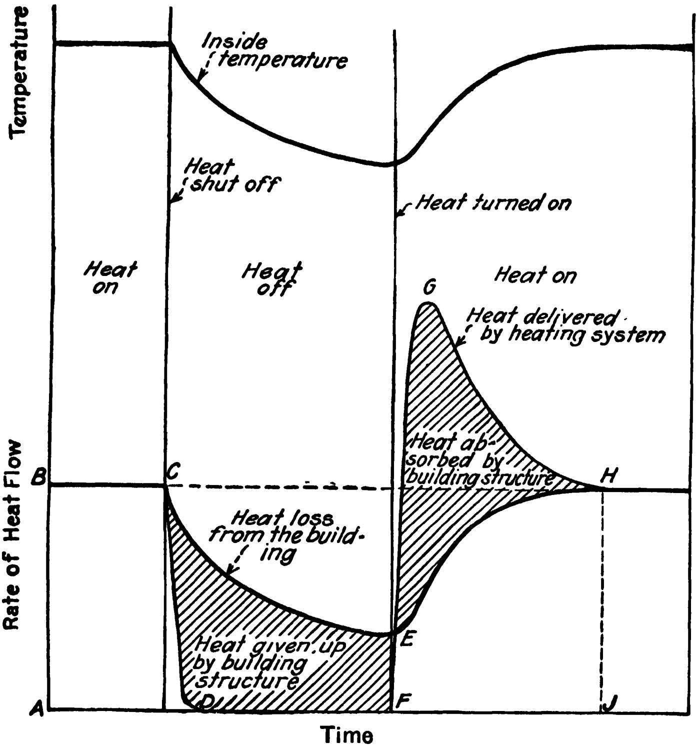

Temperature
Rate of Heat Flow
Inside temperature
Heat shut off
Heat turned on
Heat on
Heat off
Heat on
G
Heat delivered by heating system
Heat absorbed by building structure
B
C
H
Heat loss from the building
Heat given up by building structure
D
E
A
F
J
Time

FIG. 14.

At the time indicated by point *F* the heat is turned on. The rate of heat delivery increases rapidly to point *G* but not instantaneously, because the steam requires some time to fill the pipes and radiators. (A steam heating system is assumed throughout, but the same principle would hold for other systems.) For some time the rate of heat delivery is above normal, because the lower room temperature causes an increased heat output from the radiators, but it decreases until at *H* the building is up to normal temperature, and the rate of heat delivery by the heating system is equal to the rate of loss from the building. The area *EGH* represents the heat absorbed by the building structure and is equal to the area *CDFE*.

Other variables, such as the variation in outside temperature and the effect of the opening of windows and the shutting off of radiators, are omitted from this analysis, but it serves to demonstrate the principle involved. The amount of heat saved by shutting off the supply is evidently equal to the area *CEH*. There is an increased consumption of heat during the warming-up period, but it is ordinarily outweighed by the saving made during the period when the heat is off. To operate a heating system with maximum economy through shutting it off at night, the boiler capacity and the amount of radiator surface must be sufficient for warming up in a reasonable time. Otherwise the heat loss during the warming-up period, represented by area *FEHJ*, will be unduly large. The ideal condition from the standpoint of economy would be the ability to warm the building instantaneously at the beginning of the period of occupancy.

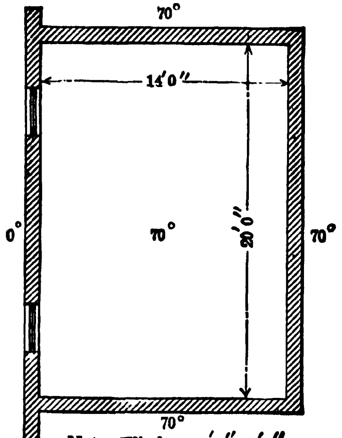
43. Heat Given Out by Persons and Processes.—In considering the amount of heat necessary to heat a room, attention must be given to the amount of heat that will be given off by the occupants of the room or by the processes that go on in it. But these sources of heat cannot always be depended upon, as it may sometimes be necessary to heat a room when there are no people in it or when the processes ordinarily going on are not in operation. On the other hand, it may be necessary to cool the room instead of heating it. Often, in a large auditorium, the greatest source of heat is the people in it. Table 14 shows the heat given off by the human body under various conditions in a room at a temperature of 70°.

The heat from electric lights is sometimes a factor, although it is more important when a room is to be artificially cooled than in the design of the heating system. The heat emitted from electric lamps is equal to the wattage of the lamps × 3.413 B.t.u per hr. It is seldom safe to depend upon this source of heat, however.

TABLE 14

	B.t.u. per Hour
Adults at rest	440
Adults at work	450– 600
Adults at violent exercise	600–1200
Children	240
Infants	63

44. Computation of Heat Losses.—It is now possible to calculate the entire heat loss from a structure. In order that the amount of radiator surface in each room may be determined, it is necessary to figure the building room by room. The total heat loss from a room is obtained by summation of the losses through the walls [H in Eq. (5), Art. 25], the losses through roofs, floors, glass, etc., similarly computed, and the infiltration losses H_s and H_l (Arts. 34 to 38) with suitable allowances and correction factors; from this total is deducted any heat produced in the room.



Note: Windows 2'-6" x 6'-0"

FIG. 15.

It should be evident that the foregoing method is not exact. It is to some extent empirical, it is necessarily based on certain assumptions, and it neglects certain factors. However, it is sufficiently accurate for ordinary purposes.

Example 1.—Assume a room as shown in Fig. 15. Let the temperature be maintained in the room at 70° and let the temperature of the outside air be 0°. Let the walls be of brick, 12 in. thick, metal lath and plaster furred on the inside; the windows, $2\frac{1}{2}$ by 6 ft.; the ceiling of the room, 10 ft. high. Let the room be on the second floor of the building, the rooms above and below heated. The windows are double-hung, wood-sash, weather-stripped.

The window openings are $2 \times 2\frac{1}{2} \times 6 = 30$ sq. ft. The gross exposed wall area is $20 \times 10 = 200$ sq. ft. The net wall area is $200 - 30 = 170$ sq. ft. From Table 6 in the Appendix the proper value of U for the wall is 0.25. The heat loss through the net wall area then $= 170 \times 0.25 \times (70 - 0) = 2975$ B.t.u.

The glass loss $= 30 \times 1.13 \times (70 - 0) = 2373$ B.t.u.

The amount of infiltration is computed as follows: The total length of crack (including the meeting rail) $(6 + 6 + 2\frac{1}{2} + 2\frac{1}{2} + 2\frac{1}{2}) \times 2 = 39$ ft.

$$39 \times 23.6 \text{ cu. ft. per hour} = 920 \text{ cu. ft. per hr.}$$

By the air-change calculation, the minimum of 0.5 of one air change per hour gives 1,400 cu. ft. per hr., which should be the figure used.

$$\frac{1{,}400(70 - 0)}{55.2} = 1776 \text{ B.t.u.}$$

The total heat loss $= 2975 + 2373 + 1776 = 7124$ B.t.u. per hr.

45. Monthly and Seasonal Heat Requirements.—The foregoing pages have been devoted to the calculation of the *maximum* heat losses. It is also frequently necessary to calculate the amount of heat consumed over a period of time. To do this it is necessary to take into account the *average* outside and inside temperatures.

It is possible to calculate these temperature differences by the hour, but an approximation is to use the unit known as the **degree-day,** which is based upon temperature difference and time. For any one day there exist as many degree-days as there are degrees Fahrenheit difference in temperature between the mean temperature for the day and 65° when the temperature is less than 65°. This may be expressed as follows:

$$\text{No. of degree-days} = (\text{ave. } t - \text{ave. } t_o) \times \text{No. of days} \quad (14)$$

It has been found that an average value for t for most types of buildings is 65°. The value of t_o may be taken as the Weather Bureau "mean temperature," which for any month is the average of the daily mean temperatures (that is, the mean between the highest and lowest readings) and is nearly equal to the true average temperature.

In order to apply the degree-day method for predicting heat consumption it is necessary to know the consumption per degree-day for the building in question or for similar buildings. Table 15 gives the steam consumption of different types of buildings expressed in pounds of steam per degree-day per thousand cubic feet of heated space. Thus for an office building having 1,200,000 cu. ft. of heated space in a city in which the heating season of 240 days had an average temperature of 40°, the seasonal steam consumption would be

$$240 \times (65 - 40) \times \frac{1{,}200{,}000}{1{,}000} \times 0.975 = 7{,}020{,}000 \text{ lb.}$$

It is not strictly correct to use average monthly or seasonal temperatures in computing degree-days, because temperatures above the base (65°) enter into the monthly averages but should not be counted in degree-day computations. The degree-days should really be calculated for each day, but as an approximation or when the daily temperatures are not obtainable the monthly or seasonal method is permissible.

TABLE 15.—STEAM CONSUMPTION PER DEGREE-DAY FOR VARIOUS TYPES OF BUILDINGS[1,2]

Type of Building	Steam Consumption, Lb. per Degree-day per Thousand Cu. Ft. of Heated Space
Apartment buildings	1.78
Hotels	1.46
Residences	1.32
Printing establishments	1.26
Clubs and lodges	0.96
Retail stores	0.90
Theaters	0.90
Loft and manufacturing buildings	0.89
Banks	0.88
Auto sales and service	0.83
Churches	0.58
Department stores	0.57
Garages (storage)[3]	0.42
Office buildings	0.975

[1] From WALKER and TUTTLE, "The Heat Requirements of Buildings," *Trans. A.S.H. & V.E.*, 1935.

[2] Includes steam for heating domestic water for heating season only.

[3] For storage garages the degree-days should be reckoned from 55° assumed inside temperature; all others from 65°.

The Weather Bureau has analyzed daily mean temperatures over a 43-year period and has established daily average degree-day figures. These daily values were then added to obtain monthly averages, and the yearly or seasonal degree-day figures given in Table 16 were established by taking a summation

TABLE 16.—AVERAGE DEGREE-DAYS FOR VARIOUS CITIES

City	Average yearly degree-days	City	Average yearly degree-days
Atlanta	3,002	New Orleans	1,208
Boston	5,943	New York	5,306
Chicago	6,287	Philadelphia	4,749
Cleveland	6,171	Portland, Oreg	4,379
Denver	5,863	San Francisco	3,143
Detroit	6,580	St. Louis	4,610
Minneapolis	7,989	Washington, D.C	4,598

of the monthly values. These data are useful in comparing various periods of fuel consumption with the averages prevailing for comparable periods.

The degree-day method should not be used to calculate heat requirements for spring or fall months only. The use of heat during those months is erratic and governed by factors other than the outside temperature only.

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Problems

1. Compute the value of U for a wall consisting of 2-in. pine boards, with still air on one side and a 20-mile wind on the other.

2. Compute the value of U for a wall similar to that shown in Fig. 11, except that wood lath and plaster are used instead of the corkboard and plaster.

3. Compute the value of U for a wall consisting of two layers of 1-in. pine boards with an air space between. Assume still air on one side and a 20-mile wind on the other. Compare with the result obtained for Prob. 1, and compute the percentage of the heat loss saved by the use of the air space.

4. The room of Fig. 15 (Example 1) requires each winter 1.3 tons of coal to heat it. How much coal would be saved if double windows were used?

5. Taking the same room as in Example 1, heated to a temperature of 60° with the surrounding rooms at 80° except those above and below which are at 60°, and the air outside −5°, how much heat must be supplied per hour?

Inside walls are of two thicknesses of wood lath and plaster with air space formed by studding. Ceiling is of wood lath and plaster, with joists and 1-in. wood floor above, and the room itself has a 1-in. wood floor with ceiling below, similarly plastered. Use values from the Appendix.

6. Given a room temperature of 74°, an outside temperature of 10°, a 10-mile wind, and a U value of 0.28, what are the outside and inside wall surface temperatures? Outside surface is brick; inside surface, smooth plaster.

7. What should be the U value of a wall that would have an inside surface temperature of 64° when the room temperature is 70° and the outdoor temperature 0°? Wall has a smooth plaster surface.

8. In the room used in Example 1 (Art. 44) compute the respective percentages of heat saving for the entire room as compared with the original construction if corkboard of ½, 1, and 1½ in. is used, with ½ in. of plaster applied directly to the corkboard. Assume 15-mile wind.

9. Assume a wall 18 in. thick of a porous material penetrable by moisture and having a conductivity of 9.0. The outdoor conditions are −10° with a 15-mile wind. Indoor conditions 73° and 40 per cent relative humidity. How far from the outside surface will condensation begin to form?

10. For the wall in Fig. 11, if the outside temperature is 10° with a 5-mile wind and the inside temperature is 72°, at what relative humidity in the room will condensation form at any point in the air space, assuming the construction to have no vapor barrier?

11. Work out Prob. 10 assuming ¾-in. wood lath and plaster instead of the corkboard and plaster.

12. Compute the gallons of water that must be evaporated per hour to maintain a relative humidity of 30 per cent with an air temperature of 72° in a residence having an internal volume of 12,000 cu. ft. Assume that air leakage is equivalent to one change per hour and that the outdoor air conditions are 20° dry-bulb temperature and 100 per cent relative humidity.

13. A building used as a retail store has 347,000 cu. ft. of heated space. The average monthly temperatures are as follows:

	Degrees		Degrees
October	51	February	22
November	42	March	39
December	35	April	48
January	20	May	63

The boiler delivers 7 lb. of steam per lb. of coal. How many tons of coal per heating season would be required?

14. Calculate the percentage increase of steam consumption that should be required to heat a St. Louis bank building having 100,000 cu. ft. of space if the total degree-days for a given year are 4,900 as compared with an average season of operation.

CHAPTER III

METHODS OF HEATING

46. Relation of Heating to Air Conditioning.—Air conditioning is defined as the simultaneous control of temperature, humidity, motion, and several other factors of the atmosphere. Under this definition, heating is a phase of air conditioning. From a practical standpoint, however, the heating of a building is often treated as a separate problem, and this is true even in some completely air-conditioned buildings. Therefore in these chapters various systems of heating will be considered, and later it will be shown how they are combined with the other functions of air conditioning.

Although in recent years greater emphasis has been placed on those other phases, there is still room and need for progress in the art of heating. In fact, a properly designed and properly controlled heating system is the major requirement for satisfactory air conditioning during the greater part of the year; and the attention now being directed at the entire problem calls for further progress in the design of the heating system.

It is desirable at this point to examine briefly the various kinds of heating systems, their fundamental principles, and their relative advantages.

47. The Evolution of the Heating System.—The most primitive method of heating was the open fire, and the first notable step in the evolution of the heating system consisted of the addition of a chimney to carry away the products of combustion, from which it was but a short step to the fireplace or grate. Heating by means of the open fire seems, through its centuries of use, to have left an indelible impression on the human mind, for the love of it still persists in spite of its serious imperfections as measured by recognized standards of what a heating system should do.

The fireplace heats a room almost entirely by radiant heat and does not directly warm the air of the room, although the air is eventually warmed by contact with the walls and contents of the room as they absorb radiant heat and by contact with the warm chimney. The heating effect on the human body is extremely uneven and erratic, and the chimney creates uncomfortable drafts; nevertheless the psychological and decorative value of the fireplace has perpetuated it to such an extent that it is retained in residences equipped with the most modern heating systems. In fact, in the British Isles and parts of Europe where the climate is less severe than in the northern United States, the fireplace is only very gradually being supplanted as the sole means of heating.

From the standpoint of thermal efficiency a fireplace ranks very low. In most cases only about 20 per cent of the heat in the fuel is effective in heating the room.

The stove, the next step in heating-system design, had certain advantages over the fireplace, but it is now seldom installed in American homes except those of cheaper types.

The warm-air furnace system and later the steam and hot-water systems were developed to satisfy the need for removing the combustion process from the occupied rooms and centralizing it in a more appropriate location. This was obviously an extremely important step, and it introduced the problem of heat distribution.

48. The Function of a Heating System.—To maintain the rooms of a building at a comfortable temperature it is necessary to supply continuously a definite amount of heat to each room, equal to the amount being dissipated from the room. It is the function of the modern heating system, taken as a whole, to extract heat from the fuel by combustion and to deliver it to the rooms in the required amounts. In the warm-air, steam, or hot-water systems the air, steam, or water should be thought of as the vehicle by which the heat is conveyed to its points of use. These three kinds of heating systems all incorporate certain essential elements, which are (*a*) a fluid for conveying the heat, (*b*) a combustion chamber in which fuel is burned and from which heat is transferred to the vehicular fluid, (*c*) pipes for conveying the fluid to the rooms, and (*d*) a means of disseminating the heat in the rooms. This last requirement is obviously simplest in the case of the warm-air system, in which the heat-conveying air becomes part of the atmosphere of the room.

49. The Warm-air Furnace.—In the warm-air furnace system the heat is distributed by air, which is taken from the rooms or partially from out-of-doors, passed over the exterior of the combustion chamber and flue-gas passages, and carried through pipes to the rooms. Cool air is taken from one or several points in the house and returned to the furnace for reheating. There is a simple type of warm-air furnace, called the "pipeless" furnace, from which the heated air is delivered only to the room directly above the furnace and passes into the other rooms through open doorways by natural circulation only. Such a system is not suitable, however, for any but the smallest houses or for small buildings having a single room.

The circulation of air in the ordinary warm-air furnace is due to the force of gravity. The difference in weight between the warm air in the pipes and the cooler air in the house causes a continuous flow of air through the system. The motive force produced by this pressure difference is exceedingly small, and the frictional resistance in the system must be made correspondingly small. This gravity-type warm-air system is also sensitive to strong outside winds, which cause excessive inleakage to the rooms on the windward side, opposing the air flow from the furnace. Notwithstanding these inherent difficulties, the gravity warm-air system has been successfully developed and is an acceptable method of heating houses of moderate size. One of its advantages is the ease with which humidification can be accomplished by the evaporation of water into the air as it circulates through the furnace.

A more recent development is the warm-air system using a fan rather than the force of gravity to produce circulation. The motive force produced by a properly designed fan is far greater than that in a gravity system, and the greatest inherent difficulty of the warm-air system is thus overcome. The use of a fan also makes it possible to filter the air and remove dust and dirt, and the system can be readily adapted to cooling in summer. At present, a great deal of attention is being given to the development of the mechanical warm-air system for residences.

50. Steam Heating.—Steam is a convenient medium for conveying heat because it condenses and gives up its latent heat of evaporation at a temperature considerably above room temperatures. A relatively large amount of heat can therefore be carried by a small amount of the circulating fluid. Another advantage is that pressure ample to produce a positive circulation can readily be obtained.

The steam system consists of a boiler in which the steam is generated, and heating units in each room which are commonly, though somewhat erroneously, called "radiators," with piping to convey the steam to them and to return the water of condensation to the boiler.

The steam system is well adapted to large buildings because of its positive circulation, its comparative cheapness of installation, and its ability to deliver large quantities of heat. Nearly all large office buildings, stores, hotels, and industrial buildings are so heated, and it is also used in many residences.

One inherent disadvantage of the steam system is its lack of flexibility. The ordinary steam radiator, unless provided with special control equipment, must either deliver its full heat output or be shut off and deliver no heat at all, because the temperature of steam cannot be varied over a wide range. The heating requirements of a room, on the other hand, vary greatly as the outside temperature changes, and this lack of coordination often results in serious temperature variations in the room. Methods have been devised of at least partially overcoming this fault, as will be discussed in later chapters, but it is a fundamental handicap of the steam system.

51. Hot-water Heating.—When water is used as the heat-carrying fluid, it gives up a portion of its sensible heat in the radiators and then returns to the heating apparatus for reheating. Since the heat thus conveyed by each pound of water is only about one-fiftieth of the heat conveyed by a pound of steam, it is necessary to circulate about fifty times as much water as steam by weight (although only one-thirtieth as much by volume).

The hot-water system is usually, though not necessarily, designed to operate at temperatures below that of the ordinary steam system, and the amount of radiator surface required is correspondingly greater. A temperature of 200° is usually the maximum unless the system is designed for high-pressure operation, in which case temperatures well above the atmospheric boiling point can be attained. In general, however, lower temperatures provide more comfortable conditions, and for that reason the authors recommend a design temperature of 160° or thereabouts.

Usually the circulation in the hot-water system is produced entirely by the difference in density of the water flowing to the radiators and the cooler water returning to the heater. This motive force is small, and the pipe sizes must be carefully chosen so as not to restrict the circulation unduly. In large buildings adequate circulation and economy in regard to pipe sizes are assured by the addition of a circulating pump. In residence heating, also, forced circulation is increasing in popularity.

The inherent advantage of the hot-water system is its ability to deliver heat at any desired rate from zero to its maximum capacity. This is accomplished by controlling the temperature of the water with respect to the outdoor temperature, and reasonably good coordination between the heat input to the rooms and their heat requirements can thus be obtained. In the past, an objection to the hot-water system was the lack of adequate provision for thus controlling the water temperature, particularly in residence systems using solid fuels. If the water temperature is not reduced on mild days, the large amount of heat stored in the water will inevitably cause overheating. Means of controlling the water temperatures are now available with various automatic fuel-burning equipment, which makes the hot-water system an excellent method of heating.

52. Fan Systems of Heating.—In buildings of a public or semipublic character, when a large number of people are gathered in a relatively small space, it is necessary to provide ventilation. This cannot be accomplished satisfactorily by merely admitting cold outdoor air; the air must first be warmed to a comfortable temperature. The ventilation therefore becomes a part of the heating problem.

There are several ways in which a combined heating and ventilating system may be arranged. One common method is to install radiators to take care of the heat losses through the walls and windows, with a fan system to supply sufficient air, warmed but little above room temperature, for ventilation. Another method is to introduce the air to the rooms at a higher temperature, thus supplying part of the heating requirements and making it possible to reduce correspondingly the size of the radiators. A third method is to supply the entire heating requirements from the fan system, eliminating the radiators.

When the fan system supplies part or all of the heating requirements, it is necessary, for proper temperature control, to make provision for varying the temperature of the air supplied to the individual rooms. This requires a more elaborate system than that used merely for supplying air at a uniform temperature for ventilating alone. One method is to install individual ducts to the various rooms, with provision for varying the temperature of the air in each duct in response to the thermostat located in the room that it supplies. Another way is to install a double set of ducts carrying air at two different temperatures. The air for the room is taken partly from each duct through a mixing damper controlled by the room thermostat.

In some industrial buildings a fan system is used for heating only, little or no outside air being introduced. Such a system has the advantage of cheapness and of effectiveness in heating large spaces.

In fan systems for heating or ventilation the air is drawn over steam or hot-water coils, by which it is heated to the desired temperature, and thence, if the building is air-conditioned, through apparatus in which it is cleaned and humidified, after which it is forced by the fan through ducts to the various rooms. Return passages of some sort are provided to remove the air, some or all of which is usually conducted back to the fan for recirculation.

It will be noted that the heating coils in such a system perform a function similar to the furnace in a warm-air furnace system.

Another type of fan system that deserves mention at this point is the unit-ventilator system. It consists of units each containing a fan, a heating coil, and often an air filter. One or more units are placed in each room, thus constituting, except for the steam supply, a complete heating and ventilating unit. This method is used principally in school buildings and factories.

53. Classification of Heating Systems.

- Grate.
- Stove.
- Warm-air furnace
  - Pipe.
  - Pipeless.
  - Fan and furnace system.
- Steam
  - Pressure system.
  - Vapor system.
  - Vacuum system.

- Hot water
  - Gravity.
  - Pressure.
  - Forced circulation.
- Fan systems, steam or water
  - Ventilation by fan—heating by direct radiation.
  - Ventilation and part of heating by fan—balance by direct radiation.
  - Ventilation and heating by fan.
  - Heating by fan—no ventilation.
  - Unit ventilator system.
  - Fan and furnace system.

54. Economy of Heating Systems.—The economy of any heating system depends upon the completeness with which the heat in the fuel is effectively utilized in heating the building. The principal sources of loss and the manner in which the heat is utilized in any type of heating system are as follows:

Losses:
- Imperfect combustion.
- Sensible heat in the chimney gases.
- Combustible in the ash.
- Radiation from boiler or furnace.
- Radiation from flues or piping.
- Losses through excessive temperature in the building.

Heat utilized:
- Heat utilized in supplying the heat losses from the building.
- Heat used in warming the air required for ventilation.

Of the losses, the first three are dependent rather upon the design of the furnace or boiler than upon the type of heating system. The radiation from the boiler or furnace is partially recovered, as it serves to warm the basement and may supply some heat to the rooms above. The loss from this source is fairly constant, regardless of the amount of heat delivered by the boiler or furnace; and if a very low fire is carried, as in mild weather, it may become quite appreciable in comparison with the heat delivered. This poor efficiency at low loads is most pronounced in the case of the steam system, as it is possible to burn a considerable amount of coal without delivering any heat to the building.

The heat used to supply the heat losses from the building is of course the principal product of any heating system. A part of this heat may be considered as an unnecessary loss, however, if excessive temperatures are maintained either during the hours when the building is occupied or during the night or other times when a low temperature could be carried.

The amount of heat used for ventilation will depend upon the amount of fresh air supplied. The air introduced for ventilation is discharged from the building at room temperature, and the heat contained in this air in excess of the heat in the outside air is chargeable to ventilation. Although this item might, from the standpoint of heating only, be considered as a loss, it is the price that must be paid for good ventilation, which is essential to comfort.

The preceding table is a classification of heating systems with regard to their mechanical construction and to the fluid (air, steam, or water) used as the heat-carrying vehicle. Another fundamental classification is whether the heating effect is produced by **radiation** or **convection** or both.

The warm-air furnace and the various kinds of fan systems, including unit heaters and ventilators, give pure convection heating; that is, only the air of the room is warmed, and there are no sources of radiant heat in the room. The same is largely true of those steam and hot-water systems that use, in place of the ordinary radiator, convection units placed in flues in the walls or in metal enclosures.

The ordinary steam or hot-water radiator gives off its heat partly by convection and partly by radiation. It is usually impracticable to heat a room *entirely* by radiant heat. To do so would require that the radiant units be placed near the ceiling so that convection currents could not sweep over them and warm the air of the room. Even so, the air of the room would ultimately become somewhat warmed by contact with the surfaces receiving radiant heat.

Panel heating in the form of completely concealed distributing pipes or ducts located either in the ceiling, walls, or floor is somewhat of a compromise between pure *radiant heating* mentioned above and conventional radiator systems. These systems are designed with *large* panels having low surface temperatures; consequently they produce a heat release that is more uniformly distributed than with a conventional radiator, which emits energy from a *small* concentrated area and at a comparatively high temperature.

There is much discussion at the present time as to the relative merits, from the standpoint of comfort and health, of panel, radiation, and convection heating.

CHAPTER IV

GRAVITY WARM-AIR FURNACE HEATING

In this chapter the simpler forms of warm-air furnace systems are discussed, in which the circulation of air is produced only by the difference in density of the air in different parts of the system. The warm-air system using forced circulation is often the basis of a partial or complete air-conditioning system and is dealt with in Chap. XXI.

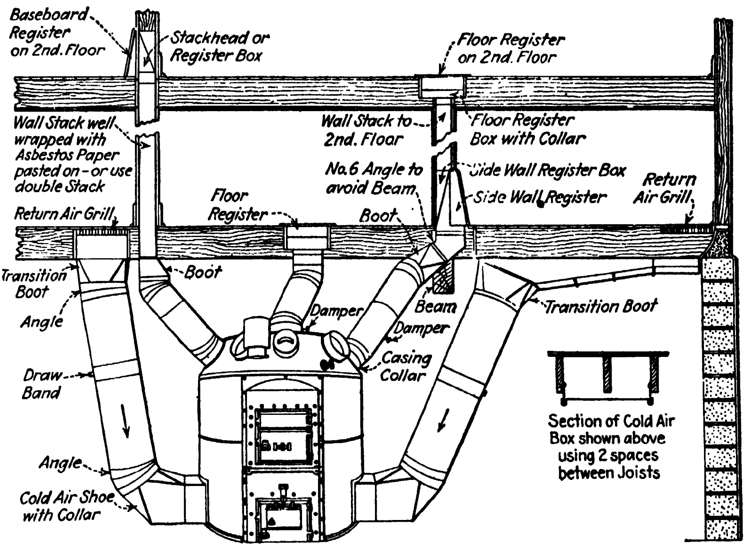

Baseboard Register on 2nd. Floor
Stackhead or Register Box
Floor Register on 2nd. Floor
Wall Stack well wrapped with Asbestos Paper pasted on - or use double Stack
Wall Stack to 2nd. Floor
Floor Register Box with Collar
No. 6 Angle to avoid Beam
Side Wall Register Box
Side Wall Register
Return Air Grill
Floor Register
Boot
Return Air Grill
Transition Boot
Boot
Angle
Damper
Beam
Damper
Transition Boot
Casing Collar
Draw Band
Section of Cold Air Box shown above using 2 spaces between Joists
Angle
Cold Air Shoe with Collar

FIG. 16.—General arrangement of furnace system. (*Waterman-Waterbury Company.*)

55. Furnace Systems.—The gravity warm-air furnace system consists essentially of the furnace, in which the heat developed from the fuel is imparted to the air, and a system of pipes to convey the warm air to the rooms and return cool air to the furnace. An exception is the so-called "pipeless system," in which the air is delivered by the furnace directly to the room above. From an engineering standpoint the pipeless furnace is relatively unimportant and will not be discussed further.

Figure 16 shows the general arrangement of an ordinary furnace system. The warm-air pipes radiate from the upper part of the furnace casing to the various rooms. The cold air enters at the bottom of the furnace. It may be taken entirely from outside or recirculated entirely from inside or taken partly from each source, depending upon the amount of fresh air that is desired for ventilation. If taken from the outside, more fuel is required, as the cold air must be heated through a greater range of temperature. For the ordinary residence it is usually unnecessary to take much, if any, fresh air, as the normal leakage of air into and out of the building is sufficient for most conditions of occupancy.

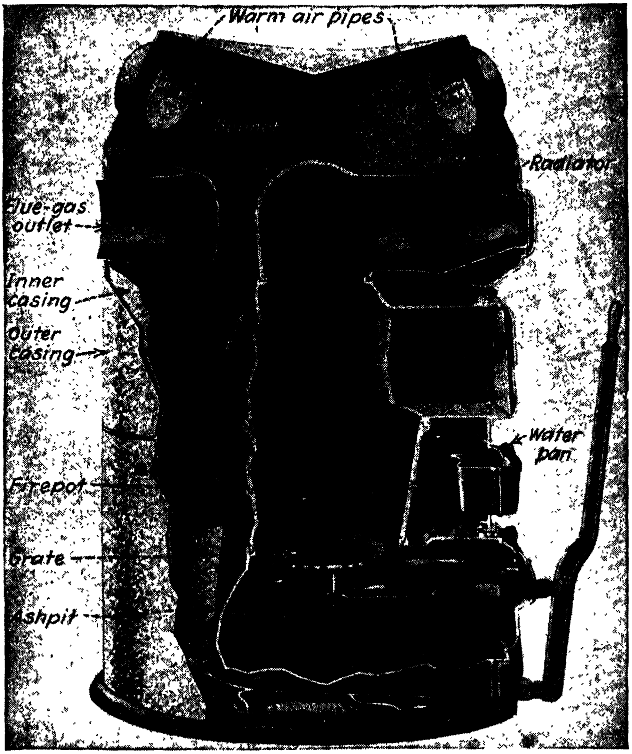


FIG. 17.—Warm-air furnace of cast-iron construction. (*Fox Furnace Company.*)

56. Furnaces.—The warm-air furnace consists fundamentally of a fire pot and a series of passages for the flue gases, surrounded by a metal casing. The air circulates through the space between the furnace proper and the casing, absorbing heat from the hot surfaces of the fire pot and gas passages. The gas passages are usually formed by a simple casting or steel heat-exchanger surface called a "radiator."

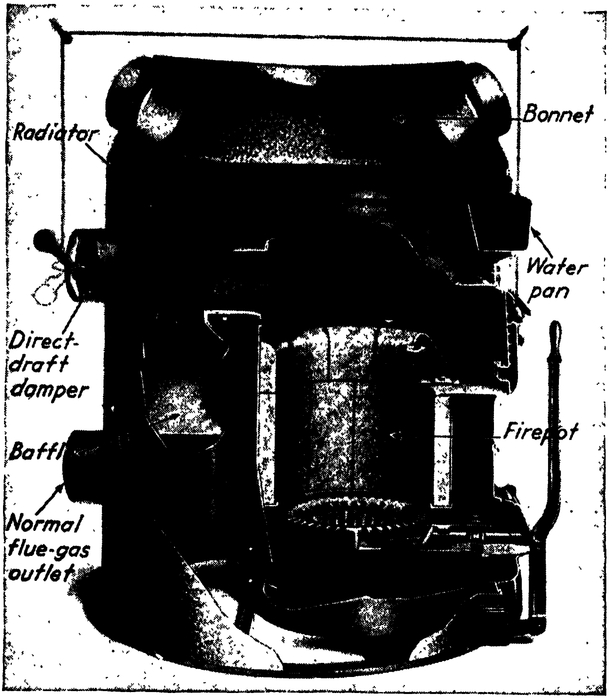
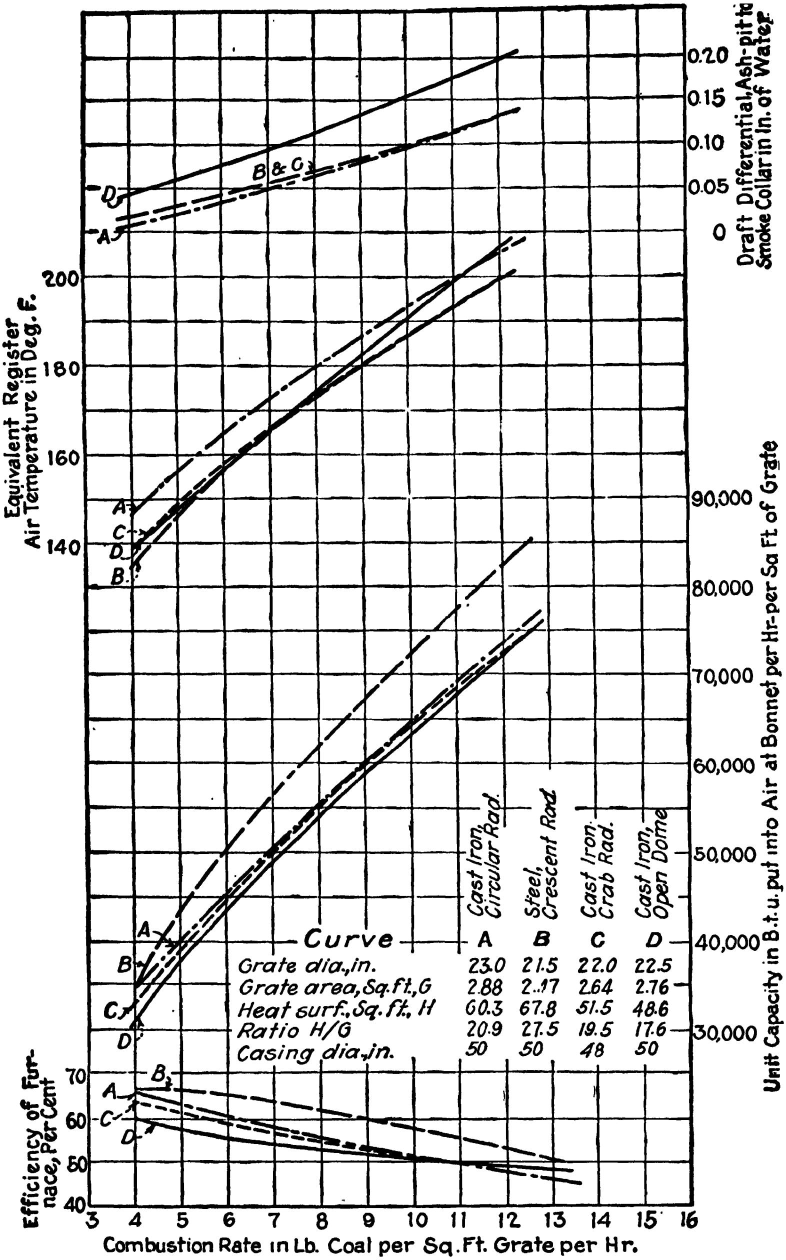


FIG. 18.—Warm-air furnace of steel construction. (*Fox Furnace Company.*)

Formerly furnaces were almost invariably made of cast iron, but in recent years the welded-steel furnace has become rather widely used. Figure 17 shows a cutaway view of a cast-iron furnace and Fig. 18 a steel furnace.

It is essential in a cast-iron furnace that the joints between the different castings be airtight so that the products of combustion cannot escape and be carried to the rooms above. The joints, therefore, are of a tongue-and-groove design, the grooves



	Cast Iron, Circular Rad.	Steel, Crescent Rad.	Cast Iron, Crab Rad.	Cast Iron, Open Dome
Curve	A	B	C	D
Grate dia., in.	23.0	21.5	22.0	22.5
Grate area, Sq. ft., G	2.88	2.17	2.64	2.76
Heat surf., Sq. ft., H	60.3	67.8	51.5	48.6
Ratio H/G	20.9	27.5	19.5	17.6
Casing dia., in.	50	50	48	50

Fig. 19.—Comparison of four types of warm-air furnaces.[1]

[1] *Bull.* 188, *Univ. Ill. Eng. Exp. Sta.*

being filled with a cement. Joints should be as few as possible, and vertical joints should be avoided.

The furnace casing is usually of galvanized steel, although large furnaces are sometimes enclosed by brickwork. When a galvanized-steel casing is used, an inner casing of black steel or tin-plated steel is provided, with an air space between the two casings for insulation.

57. Furnace Performance.—In Fig. 19 are shown the results obtained from laboratory tests on several different types of furnaces. The lower efficiency at the higher combustion rates is clearly shown. The air temperature rises rapidly as the combustion rate increases.

The efficiency results shown in Fig. 19 were obtained by considering as useful output the heat actually delivered to the circulating air. If, however, it can be considered that all of the heat developed by combustion is used, except that which escapes from the chimney top, the efficiency of the furnace is considerably higher, some tests having given results of 92 to 97 per cent with anthracite coal and an average of 75 per cent with bituminous coal.[1]

58. Humidifiers.—The warm-air furnace system affords a particularly favorable opportunity for humidification. Most furnaces have some sort of water-evaporating pan intended for this purpose, but many of them are quite ineffective in evaporating the large quantities of water required. A humidifier should have a large amount of exposed water surface located in the path of the air and should preferably receive considerable radiant or conducted heat from the furnace heating surface.

Humidifiers ordinarily fall into one of two classes: the pan type, shown in Fig. 20, and the jet type of Fig. 21. The pan type depends upon the large surface area of the water in the pan, and the jet type brings about the vaporization of the water by breaking it up into minute droplets. The pan type is automatically kept filled with water, and water is evaporated when the air in the furnace bonnet is warm. The jet type is used chiefly with furnaces equipped with a fan and is usually controlled by a humidistat, which can open the valve in the water-supply pipe only when the fan is in operation. The fan, in turn, is automatically started only when the bonnet temperature has reached a

[1] See *Univ. Ill. Eng. Exp. Sta. Bull.* 189.

predetermined value of, say, 135°. The jet-type humidifier is not very satisfactory when the supply water contains much dissolved salts. The nozzles are prone to plug, and sometimes the residue of lime or other salts is carried as a dust into the rooms.

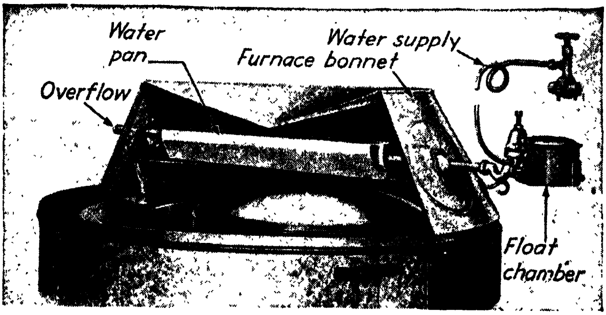


Fig. 20.—Pan-type humidifier. (*Waterman-Waterbury Company.*)

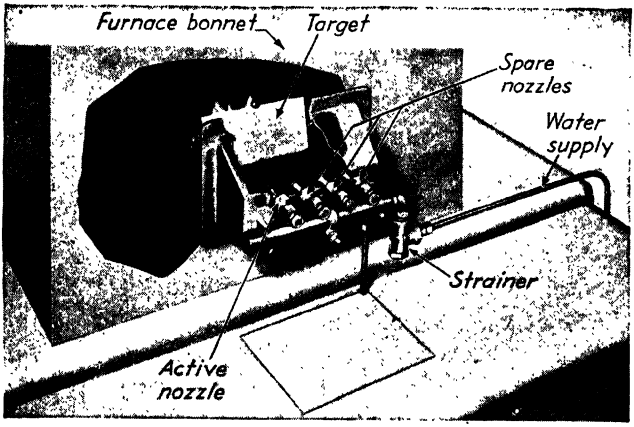


Fig. 21.—Jet-type humidifier. (*Fox Furnace Company.*)

The amount of water that must be evaporated to maintain a reasonably high humidity is greatly affected by the amount of air infiltration. If the windows are not weather-stripped, or if outdoor air is introduced through the furnace system, it is very difficult to hold a high humidity even with the humidifiers just described.

59. Warm-air Pipes.—The force producing circulation in a gravity furnace system is due merely to the difference in density of the air in the system and that of the air in the rooms or out of doors. As this force is extremely small, the design of the entire system must be such as to allow the air to circulate freely in spite of the small motive force.

The furnace has a conical hood, or bonnet, from which the warm-air pipes radiate. The horizontal pipes are called **leaders;** the vertical pipes, **stacks,** or **risers.** Leaders are usually made of round pipe. All leaders should be given the same pitch of at least 1 in. per ft., and all should leave the furnace at the same elevation.

60. The Air Flow.—Satisfactory operation of a furnace system demands the observance of a few simple principles in the piping layout, which can be readily understood upon consideration of the manner in which a furnace system operates. First of all, successful performance requires a grate of adequate size and a properly constructed chimney, because obviously the furnace itself must develop a sufficient supply of heat if the system is to operate satisfactorily. If the air pipes are ample and unrestricted, the air will circulate at a low temperature; but if the flow is restricted, then for a given total heat delivery the air must be warmed to a higher temperature not only in order to contain the given amount of heat but also in order to develop the necessary motive head for circulation. Low register temperatures are desirable because they result in more even heat distribution throughout the room and in a smaller difference in temperature between the breathing line and the floor, and because they lead to more economical operation. Therefore the leaders and stacks must be as short and .direct as possible, of ample size, and without sharp turns that would produce friction and turbulence.

The furnace should be located in a central position so that there will be no long leader pipes. The effect of a long leader pipe is different according to whether it supplies a first-floor or a second-floor register. In the former case the long leader introduces more friction and reduces the air flow, but in the latter the resistance of the stack and not that of the leader usually fixes the air flow. Consequently a long leader is not so detrimental if it supplies a second-floor register. In either case there is a marked additional heat loss from the surface of the leader pipe, which reduces the heating effect at the register, and it is therefore desirable to use heat insulation on very long leaders.

61. Cold-air Pipes.—The location of the cold-air intakes and the design of the cold-air pipes affect the operation of the system more than is generally supposed. The cold-air pipes are an important part of the air circuit, and ample capacity and freedom from turbulence are contributory to low register temperatures.

Return-air ducts should be installed from as many rooms as practicable, and from each room or series of intercommunicating rooms the air quantity withdrawn should be equivalent to the amount delivered to them.

62. Size of Warm-air Pipes.—The recommendations of the National Warm Air Heating and Air Conditioning Association are generally used in the industry for the determination of pipe sizes and furnace capacity. Based on research and experience, a physical unit *square inches of leader pipe* has been used for residences to express the heat deliveries of leader pipes, stacks, furnaces, and return ducts so as to avoid the expression **B.t.u. per hour,** which was originally thought difficult to visualize by the installer of this type of equipment. Utilizing the term **square inches of leader pipe,** it was found that 1 sq. in. of basement pipe or leader area would deliver 111 B.t.u. per hr. to a first-floor room, 167 B.t.u. to a second-floor room, or 200 B.t.u. to a third-floor room.

In general the above method has given satisfactory results, but since it was first introduced in 1922 some limitations and misapplications of the Standard Code[1] have become apparent, and in 1941 the association adopted an Application Manual,[2] in which an attempt is made to simplify the entire design procedure. Recommended combinations of leader pipes, fittings, stacks, and registers have been designated as *units,* and the proper number is obtainable from tables based on the heat loss of the room expressed in B.t.u. per hour rather than in terms of square inches of leader-pipe area.

63. N.W.A.H. and A.C.A. Standard Code.—The following outline of procedure is suggested in the Code as a method for simplifying the calculation and design of a gravity warm-air furnace system:

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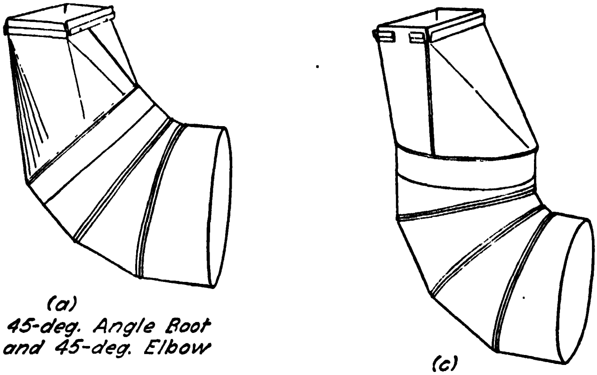
[1] The Standard Gravity Code for the Design and Installation of Gravity Warm Air Heating Systems, 11th ed., June, 1940 (N.W.A.H. and A.C.A.).

[2] Standard Code Application Manual for Gravity Warm Air Heating Systems, November, 1941 (N.W.A.H. and A.C.A.).

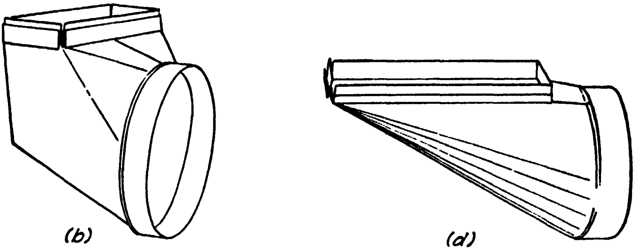
1. Determine the heat loss from each room to be heated in terms of B.t.u. per hour [as explained in Chap. II].

2. Make a tentative layout of the basement plan to scale showing proposed location of (*a*) furnace, (*b*) smoke-pipe connection to chimney, (*c*) warm-air registers (whether floor, baseboard, or wall), and (*d*) return-air grilles.



(a)
45-deg. Angle Boot
and 45-deg. Elbow

(c)
Universal Boot and
90-deg. Elbow



(b)
90-deg. Angle Boot

(d)
End Boot

Fig. 22.—Typical warm-air boots.[1]

3. Indicate on each warm-air run (*a*) whether the room to be heated is on the first or second story, (*b*) the approximate length of leader pipe in basement, and (*c*) the number of right-angle elbows as illustrated in Fig. 22. The capacity tables that follow are based on the use of the boots designated as the 90-deg. angle boot, universal boot and 90-deg. elbow, and 45-deg. angle boot and 45-deg. elbow. [If the one designated as the end boot in Fig. 22 is used, two additional elbows should be included in counting the total number of elbows; consider each 45-deg.

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[1] *Circ.* 45, *Univ. Ill. Eng. Exp. Sta.*

elbow as equal to one-half of a 90-deg. elbow; and consider the two sharp 90-deg. elbows in a crossover connection as equivalent to three 90-deg. elbows.]

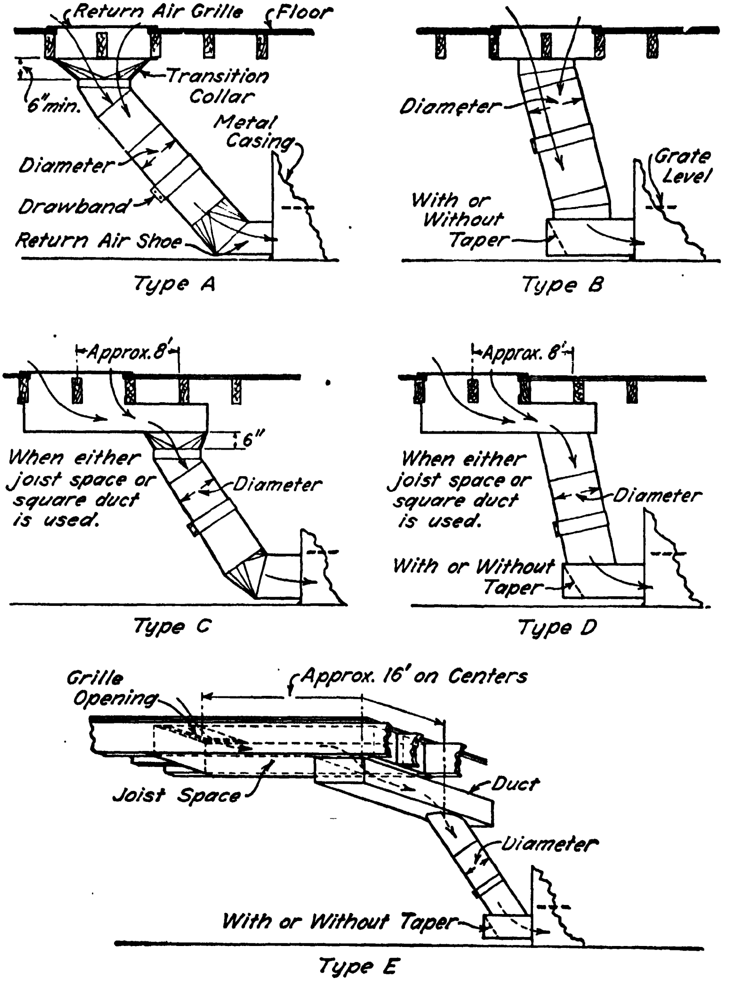

Fig. 23.—Five typical arrangements of return-air duct systems.[1]

4. Show the number and possible location of return-air grilles and the type of return-air combination as illustrated in Fig. 23.

5. The recommended combinations of warm-air and return-air pipes,

[1] *Circ.* 45, *Univ. Ill. Eng. Exp. Sta.*

ducts, fittings, registers, and grilles are designated as *units*. From Table 17 for the first story, or from Table 18 for the second story, select

TABLE 17.—WARM-AIR CARRYING CAPACITY, B.T.U. DELIVERED, FIRST-STORY REGISTERS

Unit no.	Length of leader pipe, ft.										
	4	6	8	10	12	14	16	18	20	22	24
B.t.u. Delivery with One Elbow											
1	6 020	5 850	5 680	5 510	5 340	5 170	5 000	4 830	4 660	4 490	4 320
2	7 620	7 400	7 180	6 970	6 760	6 540	6 320	6 110	5 890	5 680	5 460
3	9 400	9 140	8 870	8 600	8 340	8 070	7 810	7 540	7 270	7 010	6 740
4	13 350	12 970	12 590	12 210	11 830	11 450	11 080	10 700	10 320	9 950	9 560
5	17 520	17 020	16 530	16 040	15 550	15 050	14 550	14 050	13 560	13 060	12 560
B.t.u. Delivery with Two Elbows											
1	5 850	5 660	5 490	5 330	5 160	5 000	4 840	4 670	4 510	4 340	4 180
2	7 360	7 150	6 940	6 730	6 520	6 320	6 110	5 910	5 700	5 500	5 290
3	9 090	8 840	8 580	8 320	8 060	7 800	7 550	7 290	7 040	6 780	6 520
4	12 910	12 540	12 170	11 800	11 430	11 060	10 690	10 320	9 950	9 580	9 210
5	16 940	16 450	15 990	15 500	15 040	14 550	14 080	13 600	13 120	12 650	12 150
B.t.u. Delivery with Three Elbows											
1	5 620	5 460	5 310	5 150	4 990	4 830	4 670	4 510	4 350	4 190	4 030
2	7 120	6 910	6 710	6 510	6 310	6 110	5 900	5 700	5 500	5 300	5 100
3	8 780	8 530	8 280	8 030	7 780	7 530	7 290	7 040	6 800	6 550	6 300
4	12 450	12 100	11 750	11 400	11 050	10 700	10 350	10 000	9 650	9 300	8 950
5	16 360	15 900	15 440	14 970	14 510	14 050	13 600	13 130	12 660	12 200	11 750
B.t.u. Delivery with Four Elbows											
1	5 420	5 260	5 110	4 960	4 800	4 650	4 500	4 350	4 190	4 040	3 890
2	6 860	6 660	6 460	6 270	6 080	5 890	5 690	5 500	5 300	5 110	4 910
3	8 460	8 220	7 980	7 740	7 500	7 260	7 020	6 780	6 550	6 310	6 070
4	12 010	11 670	11 330	10 990	10 650	10 310	9 970	9 630	9 290	8 950	8 610
5	15 770	15 320	14 880	14 420	13 990	13 540	13 100	12 650	12 200	11 750	11 310
B.t.u. Delivery with Five Elbows											
1	5 240	5 090	4 940	4 790	4 640	4 500	4 350	4 200	4 050	3 910	3 760
2	6 630	6 440	6 250	6 060	5 880	5 690	5 500	5 320	5 130	4 940	4 750
3	8 180	7 950	7 720	7 490	7 260	7 030	6 800	6 560	6 330	6 100	5 860
4	11 610	11 290	10 950	10 620	10 300	9 970	9 640	9 320	8 990	8 660	8 320
5	15 250	14 800	14 380	13 950	13 520	13 090	12 650	12 230	11 800	11 370	10 940

NOTES.—1. Table 17 applies to baseboard and floor locations of warm-air registers.

2. Consider each 45-deg. elbow as equal to one-half of a 90-deg. elbow.

3. Consider each 90-deg. offset between boot and stack head equal to two 90-deg. round elbows.

4. With an end boot, include two additional elbows.

TABLE 18.—WARM-AIR CARRYING CAPACITY, B.T.U. DELIVERED, SECOND-STORY REGISTERS

Unit no.	Length of leader pipe, ft.										
	4	6	8	10	12	14	16	18	20	22	24
B.t.u. Delivered with One Elbow											
11-22	8 370	8 140	7 900	7 670	7 430	7 190	6 950	6 710	6 470	6 240	6 000
12-24	10 040	9 760	9 470	9 190	8 900	8 620	8 330	8 050	7 770	7 480	7 200
13	10 880	10 570	10 260	9 960	9 650	9 340	9 030	8 730	8 420	8 110	7 800
14	11 710	11 380	11 050	10 720	10 390	10 060	9 720	9 390	9 060	8 730	8 400
15	16 200	15 750	15 300	14 840	14 380	13 920	13 460	13 000	12 550	12 100	11 640
16	18 920	18 390	17 850	17 310	16 780	16 240	15 710	15 180	14 640	14 100	13 570
B.t.u. Delivery with Two Elbows											
11-22	7 940	7 720	7 500	7 280	7 050	6 830	6 600	6 370	6 150	5 930	5 700
12-24	9 540	9 270	9 000	8 730	8 460	8 190	7 920	7 650	7 380	7 110	6 840
13	10 340	10 050	9 750	9 460	9 170	8 880	8 580	8 290	8 000	7 700	7 410
14	11 120	10 810	10 500	10 180	9 870	9 550	9 230	8 920	8 610	8 290	7 980
15	15 400	14 970	14 530	14 100	13 670	13 230	12 800	12 360	11 930	11 500	11 070
16	17 980	17 470	16 960	16 450	15 950	15 430	14 830	14 420	13 910	13 400	12 890
B.t.u. Delivery with Three Elbows											
11-22	7 530	7 320	7 110	6 900	6 680	6 470	6 250	6 040	5 830	5 620	5 400
12-24	9 030	8 780	8 520	8 270	8 010	7 750	7 500	7 240	6 990	6 730	6 470
13	9 800	9 520	9 240	8 960	8 690	8 410	8 130	7 850	7 580	7 300	7 020
14	10 530	10 240	9 940	9 650	9 350	9 050	8 750	8 450	8 160	7 860	7 560
15	14 580	14 180	13 780	13 370	12 950	12 530	12 120	11 710	11 300	10 890	10 480
16	17 040	16 550	16 070	15 580	15 110	14 620	14 140	13 660	13 180	12 700	12 210
B.t.u. Delivery with Four Elbows											
11-22	7 120	6 920	6 720	6 520	6 310	6 110	5 900	5 700	5 500	5 300	5 100
12-24	8 530	8 290	8 050	7 810	7 570	7 330	7 080	6 940	6 600	6 360	6 120
13	9 250	8 990	8 720	8 460	8 200	7 940	7 680	7 410	7 150	6 890	6 630
14	9 950	9 670	9 390	9 110	8 830	8 550	8 260	7 980	7 700	7 420	7 140
15	13 780	13 390	13 000	12 610	12 220	11 830	11 440	11 050	10 670	10 280	9 890
16	16 080	15 620	15 170	14 710	14 260	13 810	13 350	12 900	12 440	11 980	11 530
B.t.u. Delivery with Five Elbows											
11-22	6 700	6 510	6 320	6 130	5 940	5 750	5 560	5 370	5 180	4 990	4 800
12-24	8 040	7 810	7 580	7 350	7 130	6 900	6 670	6 440	6 220	5 990	5 760
13	8 710	8 460	8 210	7 970	7 720	7 470	7 230	6 980	6 730	6 490	6 240
14	9 370	9 110	8 850	8 580	8 310	8 050	7 780	7 510	7 250	6 980	6 720
15	12 970	12 600	12 240	11 870	11 500	11 140	10 770	10 400	10 040	9 680	9 310
16	15 140	14 710	14 280	13 850	13 420	13 000	12 570	12 140	11 710	11 280	10 850

No. 21: for B.t.u. values multiply 11-22 values by 0.83.

No. 23: for B.t.u. values multiply 12-24 values by 0.83.

NOTES.—1. B.t.u. deliveries shown in Table 18 apply only to baseboard locations of warm-air registers. When floor registers are used deduct 15 per cent from the B.t.u. deliveries.

2. Consider each 45-deg. elbow as equal to one-half of a 90-deg. elbow.

3. Consider the two sharp elbows in a crossover connection as equivalent to three 90-deg. round elbows.

4. Consider each 90-deg. offset between boot and stack head equal to two 90-deg. round elbows.

5. With an end boot, include two additional elbows.

TABLE 19.—RECOMMENDED STANDARD COMMERCIAL SIZES OF LEADERS, STACKS, FITTINGS, AND REGISTERS

First-story Warm-air Ducts

Unit no.	Leader-pipe diameter, in.	Register Size, in.		
		Floor	Baseboard	
			Size	Extension*
1	8	8 × 10	10 × 8	2¼
2	9	9 × 12	12 × 8	2¼
3	10	10 × 12	12 × 9	3¼
4	12	12 × 14	13 × 11	5¼
5†	14	14 × 16		

Second-story Warm-air Ducts, Single-wall Stacks and Fittings

Unit no.	Leader-pipe diameter,	Stack‡ size, in.	Register size, in.			
			Floor	Baseboard		Side wall
				Size	Extension*	
11	8	10 × 3¼	8 × 10	10 × 8	2¼	10 × 8
12	9	12 × 3¼	9 × 12	12 × 8	2¼	12 × 8
14	10	14 × 3¼	.....	12 × 8	2¼	12 × 8
15	12	12 × 5¼	.....	12 × 10	3¼	
16	12	14 × 5¼	.....	13 × 11	3¼	

Second-story Warm-air Ducts, Double-wall Stacks and Fittings

Unit no.	Leader-pipe diameter, in.	Stack size, in.		Register size, in.			
		Internal	External	Floor	Baseboard		Side wall
					Size	Extension*	
21	8	2½§ × 10	3⅛§ × 10⅝	8 × 10	10 × 8	2¼	10 × 8
22	8	3 × 10	3⅝ × 10⅝	8 × 10	10 × 8	2¼	10 × 8
23	9	2½§ × 12	3⅛§ × 12⅝	8 × 12	12 × 8	2¼	12 × 8
24	9	3 × 12	3⅝ × 12⅝	8 × 12	12 × 8	2¼	12 × 8

* Distance from room side of plaster line to outside of register base at floor level. This allows for an increase in floor-area opening to take care of large leader-pipe diameters required and thus permits the free flow of heated air to the room with a minimum amount of restriction.

† When the calculations indicate a requirement for a given room greater than unit number 4, two or more smaller units totaling the required capacity are recommended.

‡ Recommended stack sizes. Tables may also be applied to 3-in. and 3½-in. stack depths.

§ Commercial sizes vary ⅛ in. from values shown.

the *unit* number that will supply the heat required to each room with the number of elbows and length of leader pipe previously determined. Using the *unit* number as found, read directly in Table 19 the leader, stack, and register sizes required. Double-wall stacks are sometimes used as listed in the last section of Table 19. These stacks have a double metal wall with an air space about 5⁄16 in. wide between the inner and outer walls.

TABLE 20.—RETURN-AIR CARRYING CAPACITY, B.T.U. SERVICED

Unit no.	Type A B.t.u. per hr.	Types B and C B.t.u. per hr.	Type D B.t.u. per hr.	Type E B t.u. per hr.
31	11 300	9 500	7 800	5 000
32	16 300	13 700	11 300	7 200
33	22 200	18 700	15 300	9 800
34	29 000	24 400	20 000	12 800
35	36 700	30 800	25 300	16 200
36	45 300	38 000	31 300	20 000
37	54 800	46 000	37 800	24 100
38	65 200	54 800	45 000	28 700

TABLE 21.—RECOMMENDED STANDARD COMMERCIAL SIZES OF RETURN-AIR DUCTS, FITTINGS, AND GRILLES

Unit no.	Duct diameter, in.	Area at shoe connection, sq. in.	Metal-grille sizes, in. choose one			When joist lining is used[1]		When duct is used. Choose one	
			(a)	(b)	(c)	No. of joists lined	Minimum depth,[2] in.		
31	10	. . .	. . . .	8 × 14	10 × 12	1	7	14 × 6	12 × 8
32	12	. . .	6 × 30	8 × 24	12 × 14	1	9	22 × 6	16 × 8
33	14	170	8 × 30	10 × 24	14 × 16	1	12	28 × 6	22 × 8
34	16	220	10 × 30	12 × 24	. . . .	2	8	28 × 8	22 × 10
35	18	280	12 × 30	14 × 24	. . . .	2	10	36 × 8	28 × 10
36	20	340	14 × 30	18 × 24	. . . .	2	12.5	36 × 10	30 × 12
37	22	420	18 × 30	. . . .	. . . .	2 3	15.0 10.0	42 × 10	36 × 12
38	24	500	20 × 30	. . . .	. . . .	2 3	18.0 12.0	42 × 12	36 × 14

[1] Based on 14-in. space between joists.

[2] Use full depth of joist except when joist depth is less than minimum depth required, when pan must be used.

6. From Table 20 select the *unit* number for the return-air system to correspond with the heat-loss requirements and the type of system

shown in Fig. 23. Then from Table 21 select the duct and grille size corresponding to the same *unit* number.

7. Select a furnace having a register-delivery rating, in B.t.u. per hour, equal to the total heat loss from the structure.

The Application Manual also contains detailed specifications as to furnace pipe and chimney construction and should be consulted by anyone engaged in furnace work.

Example 1.—For a sun room having a heat loss of 16,896 B.t.u. per hr., select the size of the first-story warm-air system required. There are two 90-deg. elbows and the leader is approximately 8 ft. long.

Since 16,896 B.t.u. is beyond the capacity shown in Table 17, it is necessary to select two leaders of 8448 each. From Table 17 in the 8-ft. leader column and in the section for two elbows find 8580 as the nearest capacity that corresponds to unit No. 3 in the first column. Refer to unit No. 3 in Table 19 and note that the leader should be 10 in. in diameter and should be used with a 10- by 12-in. floor register or a 12- by 9-in. baseboard register with a 3¼-in. extension.

Example 2.—Determine the size of return-air system of Type D to service 30,000 B.t.u. per hr.

From Table 20 find unit No. 36, which will service 31,300 B.t.u. per hr. Refer to Table 21 and find that unit No. 36 will require a 20-in.-diameter duct, a shoe area of 340 sq. in., a metal grille of 14 by 30 in. or 18 by 24 in., and a duct 36 by 10 in. or 30 by 12 in. If joist lining is used, the minimum depth should be 12.5 in. for two joist spaces 14 in. wide.

64. Size of Furnace.—In December, 1944, the N.W.A.H. and A.C.A. adopted a new formula for rating gravity warm-air furnaces based entirely on the heating-surface area of the furnace. This formula was derived after analyzing voluminous test data at the University of Illinois and is based on the following conditions:

a. Register air temperature not to exceed 175°.

b. Combustion rate not to be in excess of 8 lb. of coal per sq. ft. of grate area per hour.

c. Flue-gas temperature not to be in excess of 800°. (This corresponds roughly to a metal-surface temperature of about 1100°.)

d. Draft loss from ashpit to smoke collar not to exceed 0.08 in. water gage.

e. Duct transmission loss taken as 75 per cent.

f. All previously assumed safety factors have been eliminated that allowed for contingencies under service conditions, such as accumulations of soot and ashes, and ineffective firing methods.

For hand-fired furnaces converted to stoker, gas, or oil firing:

$$\text{Register delivery} = 1785 \times \text{H.S.} \tag{1}$$

For hand-fired furnaces with ratios of heating surface to grate area between 15 and 25:

$$\text{Register delivery} = 1785 \times \text{H.S.} \tag{2}$$

For hand-fired furnaces with ratios of heating surface to grate area in excess of 25:

$$\text{Register delivery} = 1785 \times 25G \tag{3}$$

where register delivery = output of furnace at register, B.t.u. per hr.
H.S. = total direct and effective indirect heating surface, sq. ft.
G = grate area, sq. ft.
Bonnet capacity = register delivery × 1.33.

Some furnace designs utilize extended fin surfaces for increasing the heat-transfer rate. The first inch of fin surface may be calculated as being 40 per cent effective in determining the rating of a unit. The second additional inch may be considered 30 per cent effective, the third additional inch as being 20 per cent effective, and for over 3 in. no allowance shall be provided in establishing the rating of the furnace.

In general, rating Eqs. (1) to (3) will reduce the former Standard Code ratings on furnaces having ratios less than about 18.5:1. It will also tend to increase the ratings for those furnaces having larger ratios, thus correcting one of the weaknesses of the old Standard Code rating, which was based on grate area and a correction factor for variations in ratio of heating surface to grate area.

The old Standard Code gave ratings expressed directly in terms of *square inches of leader-pipe area,* which may be found if desired from Eqs. (1), (2), or (3) by dividing the register delivery rating by a factor 136. This latter value is the average of the first- and second-floor leader-pipe carrying capacities referred to in Art. 62.

65. Auxiliary Fans.—The operation of a gravity warm-air furnace can often be considerably improved by the addition of a fan, provided the trouble is due to inadequate circulation. A fan cannot materially help a system having a furnace which is too small or which has an inadequate chimney.

Another frequent reason for the addition of a fan is to permit the use of a filter to remove dirt from the air.

There are available several forms of factory-built units consisting of a centrifugal fan, a filter, and often a humidifier assembled in a casing, which is installed adjacent to the furnace. One design is shown in Fig. 24.

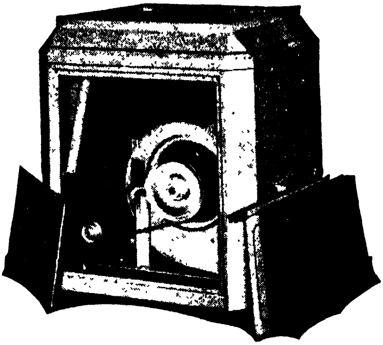

FIG. 24.—Fan unit to be attached to a gravity furnace. Filter and access panel removed. (*The Brundage Company.*)

The operation of an auxiliary fan in a gravity system radically changes the flow conditions. With gravity circulation the quantity of air that flows to any one register depends upon the frictional resistance and the gravity head; but when a fan is added, the effect of the gravity head is small as compared with the effect of the fan. Consequently, the pipes having the least frictional resistance have their flow greatly increased by the fan; and as these are usually the short first-floor leaders, the rooms that they supply receive an increased amount of heat, while the more remote rooms, which were most apt to have been underheated, may receive less heat, *proportionately*, than before; and the thermostat or the occupants may close the dampers of the furnace before the remote rooms are heated. The remedy, of course, is to adjust the dampers in the several pipes until the flow is properly balanced.[1]

Besides making damper adjustments, it is necessary, upon adding a fan to a gravity furnace, to install baffles to ensure an even flow of air around the furnace casing. Otherwise the warm air will not be properly distributed among the several leaders at the furnace bonnet. The space between the furnace proper and the casing is several inches wide, and a common method is to restrict this space, by an annular baffle, to 2 in. or thereabouts, thus distributing the air evenly around the furnace and bringing it into close contact with the heating surface.

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[1] An analysis of the effect of the fan will be found in Willard and Kratz, "Friction Losses and Observed Static Pressures in a Domestic Fan-furnace Heating System," paper presented before the A.S.H. & V.E., 1930.

The foregoing paragraphs deal merely with the auxiliary fan added to a gravity furnace. The complete mechanical warm-air furnace system is discussed in Chap. XXI.

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Problems

1. Calculate the sizes of the basement pipes, the sizes of the vertical wall stacks, and the furnace size for a residence from the following data:

Room No.	Use	Dimensions, ft.	Glass[1] area, sq. ft.	Net exposed wall area, sq. ft.	No. of sides having windows	Length leader pipe, ft.	No. elbows
Downstairs:							
1	Living room	20 × 12 × 9	65	331	3	12	3
2	Dining room	15 × 13 × 9	52	200	1	10	2
3	Kitchen	12 × 10 × 9	45	153	1	6	1
4	Hall	8 × 9 × 9	23	49	1	8	1
Upstairs:							
5	Bedroom	14 × 12 × 9	24	210	2	10	2
6	Bedroom	12 × 9 × 9	12	177	2	8	1
7	Bedroom	15 × 12 × 9	36	215	3	10	2
8	Bath	8 × 8 × 9	8	64	1	8	1

[1] That is, entire window opening.

Outside wall construction; plaster on wood lath, studding, 1 in. sheathing, building paper, 4 in. brick. This is ordinary brick veneer with no insulation. Second-floor ceiling plaster on wood lath with rough floor above. Attic not heated. Single windows not weather-stripped. Inside heated to 70°; outside temperature, 0°.

2. For the same house as in Prob. 1 calculate the basement-pipe sizes, wall-stack sizes, and furnace size, assuming the walls to be insulated with flaked gypsum fill in the studding space and double windows throughout.

CHAPTER V

RADIATORS, CONVECTORS, AND UNIT HEATERS

66. Definitions.—In a steam or hot-water heating system it is necessary to have in every room one or more heat-emitting elements to disseminate the heat conveyed by the steam or water.

Radiators and convectors are not, in fact, entirely distinct forms of apparatus, because neither gives off its heat entirely by radiation or entirely by convection. They are often collectively referred to as "radiators" or "radiation." For accuracy in designation, however, the different forms are properly termed as follows:

Direct radiators are heating units placed directly in the room to be heated.

Indirect radiators are heating units placed outside the room and delivering heat by currents of air and not as radiant heat.

Concealed radiators are radiators of the ordinary cast-iron type placed substantially within the room but largely hidden by grilles or other means.

Built-in or concealed convectors are heating units which are designed to be built into walls or partitions and which emit heat almost wholly by convection.

Cabinet convectors are heating units enclosed in a metal cabinet and set within the room.

Direct-indirect radiators are radiators set exposed in the room but designed with integral flues and arranged to take some outside air for ventilation. They are seldom used.

67. Direct Cast-iron Radiators.—Direct radiators are made of cast iron or of wrought-iron or steel pipe. The cast-iron radiator illustrated in Fig. 25 is by far the most common. It is composed of several sections cast separately and assembled, the number of sections varying according to the amount of surface required. The sections are made in several different widths and heights, so that for a radiator of any desired surface a wide range of shapes and sizes is available. The wider sections are divided through most of their length by vertical slots into two to six tubes. The standard heights vary from 14 to 32 in.

The sections are joined by means of nipples. One method uses a smooth-tapered "push nipple," fitting into tapered holes in the adjacent sections. Drawbolts extending the full length of the radiator are used to force the joints to a tight fit. Another method employs nipples threaded with right- and left-hand threads. The threaded nipples are cast with internal lugs and are turned up by means of a special wrench. The two methods of assembling are shown in Fig. 26.

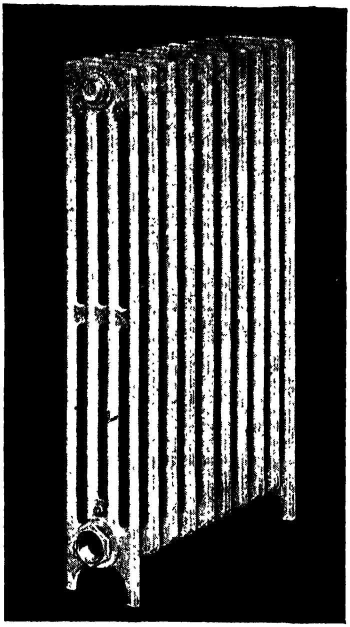

FIG. 25.—Four-tube, cast-iron radiator. (*United States Radiator Corporation.*)

The continued progress in foundry practice is reflected in the design of radiators. It has become possible to use smaller cores and thinner walls, and this is leading toward marked reduction in the over-all size and weight of radiators.

Cast-iron radiators are also furnished in the form of wall radiators, as illustrated in Fig. 27. This type of radiator is so proportioned that it takes up very little lateral space and is intended to be hung from brackets. It is well adapted for use in factory buildings.

Dimensions and ratings of small-tube radiators with a spacing of 1¾ in. per section have been standardized[1] to cover ten sizes as shown in Table 22. The large-tube type, which had a spacing of 2½ in. per section, and column radiators are no longer manufactured, but Table 1 in the Appendix lists their rated heating surface and principal dimensions.

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[1] Simplified Practice Recommendation R174-43, adopted by the Division of Simplified Practice, National Bureau of Standards, in cooperation with Institute of Boiler and Radiator Manufacturers.

Cast-iron radiators are usually given a hydraulic pressure test at the factory of about 100 lb. per sq. in. They are therefore suitable for working pressures approaching this figure but are seldom subjected to any such pressure except in the case of hot-water systems in tall buildings. The weight of the ordinary small-tube type of cast-iron radiator averages about 4.7 lb. per sq. ft. of surface, and the internal volume is about 15.5 cu. in. per sq. ft. of surface for the designs most commonly used at the present time. This internal volume is largely fixed by the requirements of manufacture, the only stipulation

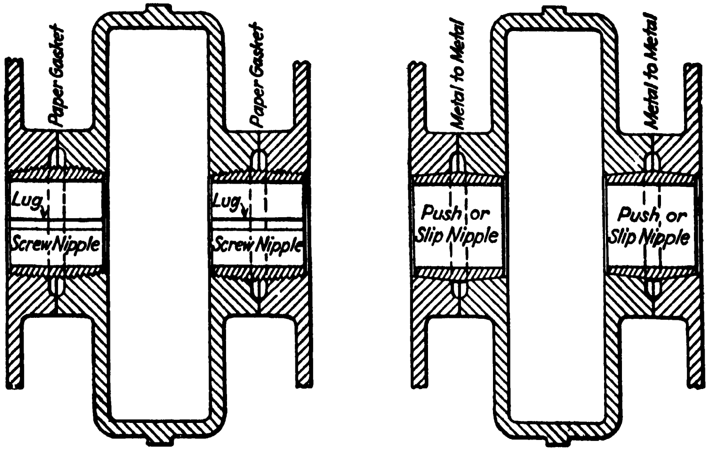


FIG. 26. Methods of assembling cast-iron radiators.

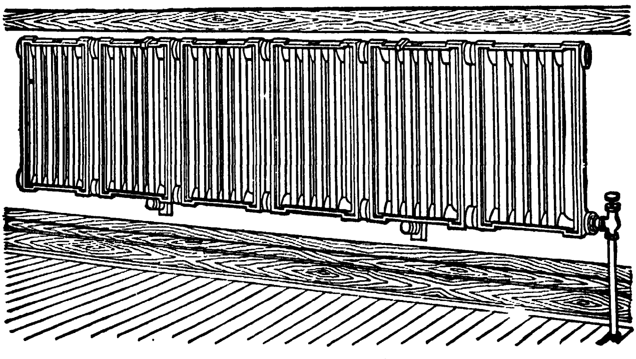
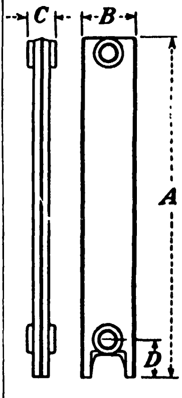


FIG. 27.—Wall radiator.

from an engineering standpoint being that the passages must not be so small as to restrict the flow of the water or steam.

TABLE 22.—SMALL-TUBE CAST-IRON RADIATORS

No. tubes per section	Catalogue rating per section*	Section dimensions				
		A Height†	B Width		C Spacing‡	D Leg height†
			Minimum	Maximum		
	Sq. ft.	In.	In.	In.	In.	In
3§	1.6	25	$3\frac{1}{4}$	$3\frac{1}{2}$	$1\frac{3}{4}$	$2\frac{1}{2}$
4§	1.6	19	$4\frac{7}{16}$	$4\frac{13}{16}$	$1\frac{3}{4}$	$2\frac{1}{2}$
	1.8	22	$4\frac{7}{16}$	$4\frac{13}{16}$	$1\frac{3}{4}$	$2\frac{1}{2}$
	2.0	25	$4\frac{7}{16}$	$4\frac{13}{16}$	$1\frac{3}{4}$	$2\frac{1}{2}$
5§	2.1	22	$5\frac{5}{8}$	$6\frac{5}{16}$	$1\frac{3}{4}$	$2\frac{1}{2}$
	2.4	25	$5\frac{5}{8}$	$6\frac{5}{16}$	$1\frac{3}{4}$	$2\frac{1}{2}$
6§	1.6	14	$6\frac{13}{16}$	8	$1\frac{3}{4}$	$2\frac{1}{2}$
	2.3	19	$6\frac{13}{16}$	8	$1\frac{3}{4}$	$2\frac{1}{2}$
	3.0	25	$6\frac{13}{16}$	8	$1\frac{3}{4}$	$2\frac{1}{2}$
	3.7	32	$6\frac{13}{16}$	8	$1\frac{3}{4}$	$2\frac{1}{2}$



radiators exposed in a normal manner; not to radiators installed behind enclosures, grilles, etc. (See A.S.H. & V.E. Code for Testing Radiators.)

† Over-all height and leg height, as produced by some manufacturers, are 1 in. greater than shown in columns A and D. Radiators may be furnished without legs. Where greater than standard leg heights are required this dimension shall be $4\frac{1}{2}$ in.

‡ Length equals number of sections times $1\frac{3}{4}$ in.

§ Or equal.

It should be noted that the height of a radiator (except wall radiators) is taken as the *total height above the floor* for radiators having legs of standard height. The rated surface given in the table does not correspond exactly with the actual surface.

68. Equivalent Radiator Surface.—An equivalent square foot of steam radiator surface is that amount of surface which, with a steam temperature of 215° and a surrounding air temperature of 70°, will emit 240 B.t.u. per hr. The use of this unit for rating radiators, as in Table 22, makes it unnecessary to know the *actual* amount of surface, which is not an accurate index of heat output even if known. The capacity of radiators and convectors of various types may therefore be expressed as equivalent direct steam-radiator surface by simply dividing their heat output, in B.t.u., under standard conditions, by 240.

In hot-water systems an equivalent square foot of radiator surface is sometimes considered as emitting 150 B.t.u. per hr. The general use of this unit is inadvisable, however, because of the different temperatures used in hot-water systems.

69. Radiator Tappings.—The end sections of cast-iron radiators are usually tapped for a 2-in. pipe thread and furnished with bushings having openings whose size depends on the size of the radiator. The sizes of the reduced openings for radiators intended for use with different systems of piping are as follows:

TABLE 23.—RADIATOR TAPPINGS

	Single-pipe Work	
Size of radiator, sq. ft.	Pipe size of tapping, in.	
Up to 24	1	
24 to 60	1¼	
Above 60	1½	
	Two-pipe Work	
	Supply	Return
Up to 48	1	¾
48 to 96	1¼	1
Above 96	1½	1¼
	Water Radiators	
	Supply	Return
Up to 40	1	1
40 to 72	1¼	1¼
Above 72	1½	1½

For vapor systems, supply, ¾ in., return, ½ in. Air-valve tapping, ⅛ in. on all radiators.

Radiators are now made according to the same design whether used for steam or for water. Formerly a different design with no upper connections between the sections was used with steam, but the manufacture of this type has been discontinued.

In industrial buildings radiators built of pipe are occasionally used and are satisfactory. These pipe coils usually consist of a pair of cast-iron headers connected by four or more pipes of either 1 or 1¼ in. diameter. Pipe coils are usually made in the miter form, as shown in Fig. 28. The vertical lengths of pipe provide sufficient flexibility to allow the longer horizontal members to expand freely. Some such provision is essential.

The openings in one of the headers or the elbows are tapped with a left-hand thread so that the coil can be readily assembled.

70. Indirect and Concealed Radiators.—**Indirect radiators** are so named because they are located outside the room to be heated, and the heat is conveyed from the radiator to the room by a current of air. The term is a general one and is even applied to fan systems of heating and ventilation. Any of the various forms of steam or water radiators or convectors, when placed outside the room and connected to it with ductwork, may be called an indirect radiator.

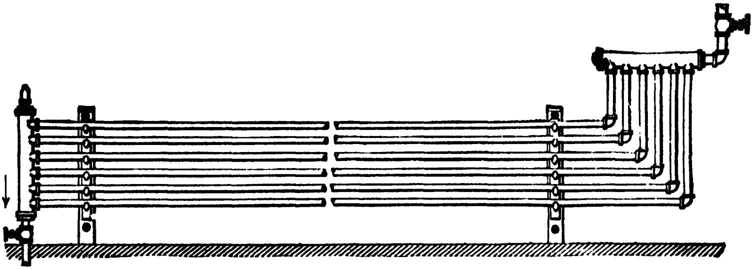


Fig. 28.—Miter pipe coil.

Concealed radiators are quite common and need no special consideration here. The form and dimensions of the enclosure, or grille, have an important effect on the performance of the radiator, as shown in Art. 80.

71. Concealed Convectors.—Several commercial forms of built-in or concealed convectors have been developed, complete with metal enclosures designed to fit into partitions or into outside walls beneath the windows. This type of heater is popular chiefly because it is virtually invisible when installed. A typical form is shown in Fig. 29.

The heating elements used in most concealed heaters are of brass or copper, consisting usually of tubes with thin fins, pressed on in some manner so as to give a greatly extended surface of metal having high heat conductivity, so that a large amount of heat is emitted from a heating element having relatively small over-all dimensions. The heating element is placed near the bottom of a stack or flue of appreciable height, which gives the required chimney effect. The heat output may be controlled by a damper placed in the air-outlet grille or in the stack.

The heating element for one design of convector is shown in Fig. 30. It consists of cast-iron headers into which the ends of three copper tubes are fastened by rolling. The three tubes, which are the **primary heating surface,** perforate a large number of closely spaced plates or fins, which are enclosed with sheet-metal side plates. The object sought is the maximum heat emission with a minimum amount of metal and of space occupied.

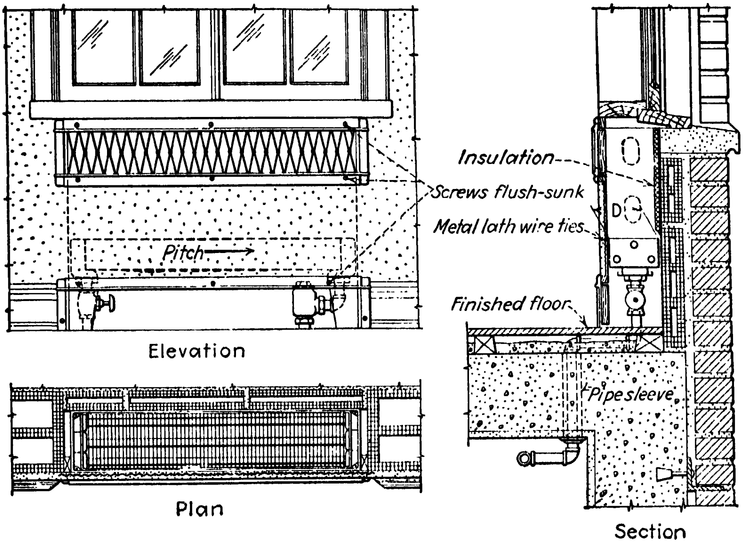


Fig. 29.—Concealed convector. (*Modine Manufacturing Company.*)

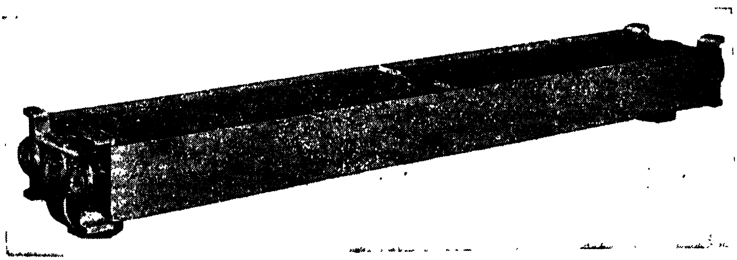


Fig. 30.—Heating element for convector. (*Trane Company.*)

Widening the fins, which constitute the **secondary heating surface,** increases the heat emission but at a decreasing rate, so that it becomes uneconomical to use extremely wide fins.

There are on the market a number of ingenious designs of nonferrous heating elements, which all embody the same basic principle of using metals of high thermal conductivity and closely spaced fins. Various cast-iron convector elements are also available.

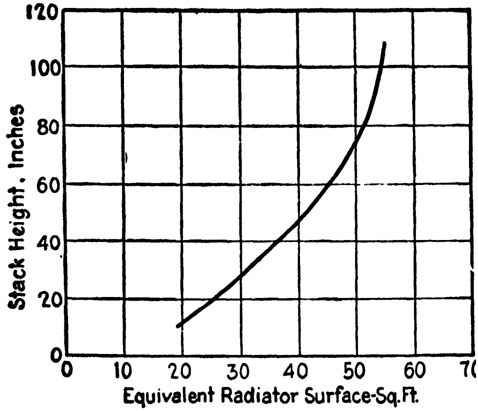
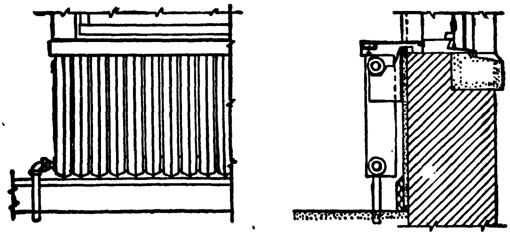


FIG. 31.—Effect of stack height on heat output of a concealed convector.[1]

The heat emission naturally depends upon the height of the flue or stack. Figure 31 illustrates the effect upon the heat output of increasing the height of the stack for one form of heating unit. Increasing the height above about 7 ft. did not increase the heat output appreciably. Also, it is undesirable for the warm air to be discharged close to the ceiling of the room because of the resulting stratification.



Elevation Vertical Section

FIG. 32.—Cast-iron, suspended, flue-type radiator.

Several forms of cast-iron radiators have been introduced to compete with the concealed heaters. One of them, designed to be hung below the window sill, is shown in Fig. 32, the sill being usually of metal and perforated to form a grille. The face of this form of radiator is exposed, and the proportion of radiant heat emitted is large as compared with that of the true concealed heaters, varying from 20 to 40 per cent of the total heat, depending upon the width.

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[1] From tests by R. N. Trane, reported in *Trans. A.S.H. & V.E.*, 1927.

Comparison of Fig. 32 with the convector unit in Fig. 29 and with the direct radiator shows the great diversity in design of heat-emitting units. The question of the comfort value of radiant heat as compared with heating by convection is at present a subject of much discussion.

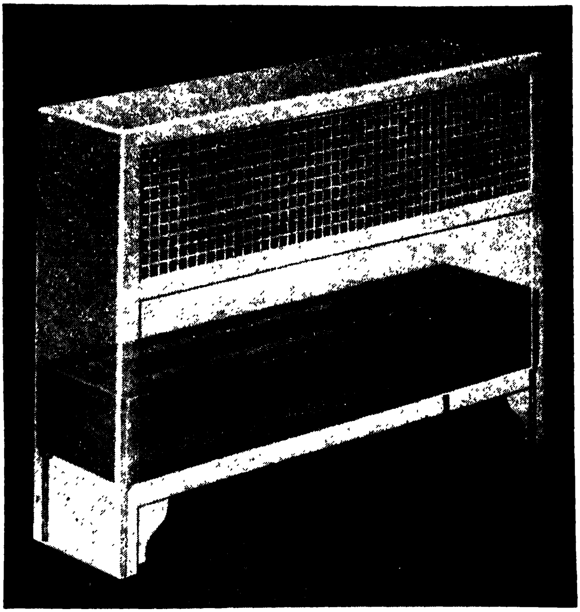

FIG. 33.—Cabinet convector. (*United States Radiator Corporation.*)

72. Cabinet Convectors.—The cabinet convector functions exactly like the concealed convector but does not need to be built into the wall and consequently is less expensive to install, particularly in existing buildings. Figure 33 shows a type in which the finned heating element is cast in one piece.

Cabinet convectors are frequently built with the outlet grille at the top instead of the front, which causes the air to be discharged vertically upward. This design is sometimes preferred when an unusual downdraft from tall windows is to be overcome.

73. Heat Emission from Radiators.—Heat passing from the water or steam in a radiator to the air of a room encounters a series of resistances to heat flow similar to those in a building wall. There is a surface resistance at the inner metal surface, a resistance to heat flow through the metal wall, and a resistance to heat emission from the outer metal surface. The first two of these items are insignificant as compared to the last, as shown by the temperature drop, which is less than 0.5° from the steam to the outer surface, as compared with, say, 145° (that is, 215 to 70°) from the outer surface to the air. In a radiator, the outer surface resistance is the neck of the bottle in regard to heat flow.

The emission of heat from the outer surface takes place by radiation and by convection. For the complex configuration of a radiator it would be impossible to determine the heat emission without recourse to laboratory tests, but the formulas that have been developed for simple bodies such as cylinders are of value in illustrating the principles involved. The radiation and convection are best treated as separate factors. The radiated heat as calculated by the well-known Stefan-Boltzmann equation is

$$H_r = 0.1723e\left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4\right] \qquad (1)$$

where H_r = heat loss by radiation, B.t.u. per sq. ft. per hr.
e = emissivity coefficient.
T_1 = temperature of hot surface, deg. abs.
T_2 = temperature of surrounding objects, deg. abs.

The emissivity coefficient e is an important quantity. It varies from 0.10 for highly polished copper to 0.94 for rough steel. The *nature of the surface* thus exerts a marked effect on the heat given off by radiation and consequently on the total heat emission from a radiator. Varying the surface finish will vary the proportions of radiant and convected heat and affect the comfort conditions in a room.

The heat given off by convection (from a cylinder) may be calculated by the Rice-Heilman equation

$$H_c = C\left(\frac{1}{D}\right)^{0.2} \times \left(\frac{1}{T_a}\right)^{0.181} \times (t_1 - t_2)^{1.266} \qquad (2)$$

where H_c = heat loss by convection, B.t.u. per sq. ft. per hr.

C = a constant depending on configuration = 1.016 for horizontal cylinders.

D = diameter, in.

T_a = average absolute temperature, deg. F., of surface and air.

$(t_1 - t_2)$ = difference in temperature between the surface and air, deg. F.

Then the total heat loss is $H_r + H_c$. Willard and Kratz calculated H_r and H_c for a 10-in. horizontal cylinder to be, respectively, 205 and 107 B.t.u. per sq. ft. per hr. which agreed closely with test results.

For radiators the relation between radiant and convected heat varies widely, depending upon the configuration. A thin, flat radiator, such as a wall radiator, gives off the major part of its heat by radiation, whereas a thick one, such as a five-tube radiator, gives off more heat by convection. A built-in or cabinet convector, of course, works almost wholly by convection.

One recent test indicates that from 40 to 50 per cent of the heat output of an ordinary radiator is radiant heat.[1]

Although the question whether Eqs. (1) and (2) actually hold true for radiators is rather academic, there is a practical need for a reasonably accurate formula for translating test conditions to actual conditions. If a radiator is tested and given a rating for the standard conditions of 215° steam and 70° room air, one must be able to determine its expected output for other steam or air temperatures. Or, vice versa, if the test is not made under standard conditions, it is necessary to use a correction factor to compute the output under standard conditions.

74. Radiator Performance.—The A.S.H. & V.E. "Code for Testing Radiators" (1927)[2] assumes that the heat output of a *radiator* is in accordance with the formula

$$H = K(t_s - t_1)^n \qquad (3)$$

---

[1] Rubert, "Heat Emission from Radiators," *Cornell Univ. Eng. Exp. Sta Bull.* 24.

[2] *Trans. A.S.H. & V.E.*, 1927.

where H = heat output, B.t.u. per hr.
K = a constant, depending on the size and shape.
t_s = temperature of the steam, deg. F.
t_1 = room-air temperature, deg. F.
n = a numerical exponent.

The value taken for n is 1.3, and the Code gives the following rule:

The heating capacity of a radiator shall be determined as follows:

$$\left.\begin{array}{l}\text{B.t.u. emission per hour at } 70^\circ \\ \text{room and } 215^\circ \text{ steam temperatures}\end{array}\right\} = \left\{\begin{array}{l}\text{Pounds of condensation per} \\ \text{hour} \times 970 \times \text{correction factor}\end{array}\right. \tag{4}$$

Correction factor C_s =

$$\left\{\frac{215 - 70}{\text{Average steam temperature} - \text{room temperature}}\right\}^{1.3} \tag{5}$$

Though this is a somewhat empirical formula, it has been verified experimentally through a fairly comprehensive temperature range.

The performance of a radiator or convector under different conditions can be compared by modifying Eq. (3) as follows:

$$\frac{H}{H'} = \left(\frac{t_s - t_1}{t_s' - t_1'}\right)^n \tag{6}$$

where n for radiators is 1.3, and
H = heat output for first condition.
t_s = steam (or average water) temperature for first condition.
t_1 = room-air temperature for first condition.

The symbols in the denominators represent corresponding quantities for the second condition.

75. Heat Output of Convectors.—In *convectors* the heat emission is fairly well expressed by Eq. (4) but with the correction factor determined by the formula

$$C_s = \left(\frac{215 - 65}{t_s - t_i}\right)^{1.5} \tag{7}$$

where C_s = correction factor, to be applied in Eq. (4).

t_s = steam temperature in test.

t_i = average inlet-air temperature in test.

The exponent 1.5 has been shown by test[1] to be the proper figure for convectors as compared with the figure of 1.3 for radiators. The value 1.3 is given for convectors in the A.S.H. & V.E. Code,[2] but in view of test results it appears to be incorrect. Equation (6) with $n = 1.5$ is also applicable to convectors.

In the case of hot-water convectors the correction factor is determined by the formula

$$C_w = \left[\frac{\theta_s - 65}{\frac{\theta_1 + \theta_2}{2} - t_i}\right]^{1.5} \qquad (8)$$

where C_w = correction factor, as before.

θ_s = one of the standard mean water temperatures 170, 190, 210, or 230°. The test is made with a 20° temperature drop in the radiator.

θ_1 = inlet-water temperature, deg. F.

θ_2 = outlet-water temperature, deg. F.

In Table 24 are given the values of the correction factors based on Eqs. (5) and (7) for steam or water radiators and convectors. When used for water, however, the factors in the table are applicable only for correcting heat outputs to a 215° base. For other temperatures Eq. (8) must be used.

Example 1.—Given a radiator-test result of 13.7 lb. of condensation per hr. for a 50-sq. ft. radiator. Room temperature during test, 75°; steam temperature, 230°. What would be the heat output per square foot under standard conditions?

The correction factor from Table 24 is 0.92. From Eq. (4)

$$\text{B.t.u. per hr. per sq. ft. under standard conditions} = \frac{13.7}{50} \times 970 \times 0.92$$
$$= 244 \text{ B.t.u.}$$

Example 2.—Given a radiator that will deliver 240 B.t.u. per hr. per sq. ft. under standard conditions. How much heat will it deliver if the steam temperature is 200° and the room temperature 80°?

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[1] KRATZ, FAHNESTOCK, and BRODERICK, "Factors Affecting the Heat Output of Convectors," *Trans. A.S.H. & V.E.*, 1934.

[2] A.S.H. & V.E. "Standard Code for Testing and Rating Concealed Gravity Type Radiation" (Steam Code), *Trans. A.S.H. & V.E.*, 1931.

The correction factor is 1.28. From Eq. (4)

$$240 = \text{B.t.u. per hr.} \times 1.28$$

$$\text{B.t.u. per hr.} = \frac{240}{1.28} = 187.5$$

TABLE 24.—CORRECTION FACTORS FOR DIRECT CAST-IRON RADIATORS AND CONVECTOR HEATERS[1]

Steam pressure, approximate			Steam or water temperature, °F.	Factors for direct cast-iron radiators							Factors for convectors						
				Room temperature, °F.							Inlet-air temperature, °F.						
	Gage	Lbs. abs.		80	75	70	65	60	55	50	80	75	70	65	60	55	50
Vacuum, in. Hg.	22.4	3.7	150	2.58	2.36	2.17	2.00	1.86	1.73	1.62	3.14	2.85	2.57	2.35	2.15	1.98	1.84
	20.3	4.7	160	2.17	2.00	1.86	1.73	1.62	1.52	1.44	2.57	2.35	2.15	1.98	1.84	1.71	1.59
	17.7	6.0	170	1.86	1.73	1.62	1.52	1.44	1.35	1.28	2.15	1.98	1.84	1.71	1.59	1.49	1.40
	14.6	7.5	180	1.62	1.52	1.44	1.35	1.28	1.21	1.15	1.84	1.71	1.59	1.49	1.40	1.32	1.24
	10.9	9.3	190	1.44	1.35	1.28	1.21	1.15	1.10	1.05	1.59	1.49	1.40	1.32	1.24	1.17	1.11
	6.5	11.5	200	1.28	1.21	1.15	1.10	1.05	1.00	0.96	1.40	1.32	1.24	1.17	1.11	1.05	1.00
Lb. per sq. in.	1	15.6	215	1.10	1.05	1.00	0.96	0.92	0.88	0.85	1.17	1.11	1.05	1.00	0.95	0.91	0.87
	6	21	230	0.96	0.92	0.88	0.85	0.81	0.78	0.76	1.00	0.95	0.91	0.87	0.83	0.79	0.76
	15	30	250	0.81	0.78	0.76	0.73	0.70	0.68	0.66	0.83	0.79	0.76	0.73	0.70	0.68	0.65
	27	42	270	0.70	0.68	0.66	0.64	0.62	0.60	0.58	0.70	0.68	0.65	0.63	0.60	0.58	0.56
	52	67	300	0.58	0.57	0.55	0.53	0.52	0.51	0.49	0.56	0.54	0.53	0.51	0.49	0.48	0.47

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[1] Factors taken from A.S.H. & V.E. "Guide," 1945.

Example 3.—Determine the heat emission rate in B.t.u. per hour per square foot for small-tube cast-iron radiators to be used in a forced hot-water heating system where the supply-water temperature is to be 210° and the return water 190°. The resultant mean water temperature in the radiator will be 200°. For a room temperature of 70°, the correction factor (Table 24) is 1.15. Using 240 as the heat emission under standard steam rating conditions

$$H = \frac{240}{1.15} = 209 \text{ B.t.u. per hr. per sq. ft.}$$

for the hot-water-circulating conditions specified.

If the designed mean water temperature had been indicated as 170°F., then the heat-emission rate would have been 240/1.62 or 148 B.t.u. per hr. per sq. ft., which is reasonably close to the value of 150 referred to in Art. 68.

76. Effect of Painting.—The practical significance of variations in the emissivity coefficient e [Eq. (1), Art. 73] is shown by the effect of different kinds of paint on the heat emission. Radiator paints are of two kinds, those containing finely ground pigment, and the bronze or aluminum paints, which contain metal in flake form. The amount of heat emitted by radiation is considerably less from surfaces covered with these flake-metal paints, the effect being entirely on the surface. Table 25 shows the results obtained in tests[1] upon a 10-section 26-in. four-tube cast-iron radiator which was painted with various kinds of paint and the performance of which was compared with that of a similar unpainted radiator.

TABLE 25.—EFFECT OF PAINTING ON HEAT EMISSION

Finish	Relative Heat Emission
Bare radiator	100.0
Aluminum paint	93.7
Gold-bronze paint	92.6
Flat brown paint	104.8
Flat cream paint	104.0
White gloss enamel	102.2

The final coat, if more than one kind of paint is applied, is the one that determines the result. It will be seen that the over-all heat emission for the kind of radiator tested may be either decreased or increased as compared with the bare iron surface. With a heating unit that gives up heat almost entirely by convection the effect would be very slight.

In ordinary practice the effect of painting is seldom taken into account in selecting the size of radiators.

77. Heat Emission from Wall Radiators and Pipe Coils.—Cast-iron wall radiators are made in three sizes containing, respectively, 5, 7, and 9 nominal square feet of surface. The heat emission under standard conditions has been found to be as follows:

TABLE 26.—HEAT EMISSION FROM CAST-IRON WALL RADIATORS

Heating surface, square feet	Long side	Total heat emission, B.t.u. per hour
5	Vertical	1390
5	Horizontal	1530
7	Vertical	1730
7	Horizontal	1950
9	Vertical	2310
9	Horizontal	2580

The heat emission from pipe-coil radiators naturally depends upon the size of pipe used and upon the arrangement of the coil. Table 27 gives accepted values.

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[1] FESSENDEN and MARIN, *Trans. A.S.H. & V.E.*, 1929.

TABLE 27.—HEAT EMISSION OF DIRECT PIPE-COIL RADIATORS FOR STEAM (Steam temperature 215°F. Room temperature 70°F.; coils placed vertical —pipes horizontal. B.t.u. per lineal feet of coil per hour [not lineal feet of pipe])

No. of rows	Size of coil		
	1 in.	1¼ in.	1½ in.
1	132	155	185
2	252	312	348
4	440	545	616
6	567	702	793
8	651	796	907
10	732	907	1020
12	812	1005	1135

NOTE.—For coils and pipes both placed vertical, use 100 B.t.u. per lin. ft. of pipe as an average for a 1¼-in. coil, 10 ft. high.

For horizontal ceiling coils emission is less than that of a single pipe when the coil is surrounded by air at a higher temperature, as is usually the case. Use

126 B.t.u. per lin. ft. of pipe for 1-in. coils.
146 B.t.u. per lin. ft. of pipe for 1¼-in. coils.
175 B.t.u. per lin. ft. of pipe for 1½-in. coils.

78. The Heating Effect of Radiators.—Thus far the function of a radiator or convector has been considered as merely that of delivering so much heat to the room. It is necessary to take into account, however, the heating effect as related to the occupants of the room. There are, in general, two points to be considered, the first being that the vertical distribution of air temperatures in the room depends upon the manner in which that portion of the heat that is given off by convection is delivered to the room. In any heated room there is a temperature gradient between the floor and ceiling because of the natural tendency of heated air to rise. Since the zone of occupancy is between the floor and the 6-ft. level, the heat that is delivered to the upper part of the room is not so effective as that which is delivered to the lower levels. Furthermore, a small temperature gradient *within* the zone of occupancy is more desirable from the comfort standpoint than a large gradient.

A radiator that warms, to a high temperature, the current of air passing upward through it causes a relatively large amount of heat to be carried toward the ceiling. A study of Fig. 34 will show how strikingly this is borne out by actual tests. The curves show the temperature gradients and the rates at which steam was condensed in the four types of radiators illustrated. The five-tube radiator, which required a length of only nine sections to heat the room, caused the greatest temperature differential, whereas the wall radiator caused the least. The condensation rates varied in an inverse manner with the uniformity of temperature. Note that in these tests a uniform temperature was maintained at the 30-in. level, which gave better results than the common method of controlling the temperature from the 5-ft., or "breathing-line," level.

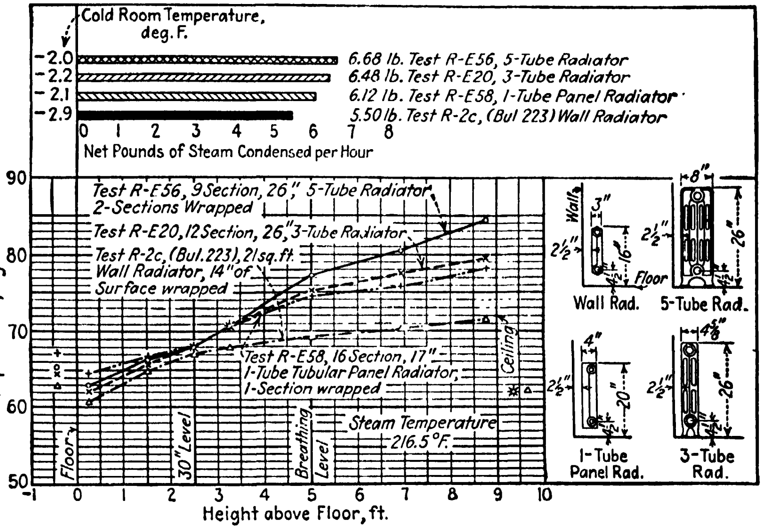

FIG. 34.—Temperature gradients produced by different forms of radiators.[1]

The **second** point to be considered regarding heating effect is that the human body receives and emits appreciable amounts of heat by radiation. The amount of heat given off by radiation from a radiator depends upon the shape of the radiator, so that two radiators of the same total heat emission but of different shapes may differ in their heating effect. The human body radiates heat to cold wall and window surfaces, and this also has a bearing on the comfort in a room. These factors are very complex and are not taken into account in selecting the *size* of the radiators but should be considered in selecting the *type* and *location* of radiators. For example, the occupants of an office or schoolroom who must sit very close to a radiator may suffer intense discomfort from the radiant heat. If ordinary radiators are used in such places, they should be provided with shields to cut down the radiant heat.

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[1] From KRATZ and FAHNESTOCK, "Steam Condensation, an Inverse Index of Heating Effect," *Trans. A.S.H. & V.E.*, 1931.

One method of evaluating the heating effect of a particular system, which has already been defined in Art. 8, Chap. I, is to establish the **mean radiant temperature** of the environment. As the term implies, this temperature is the mean value of all the radiating surfaces in an enclosure, irrespective of whether they are emitting or receiving heat by radiation.

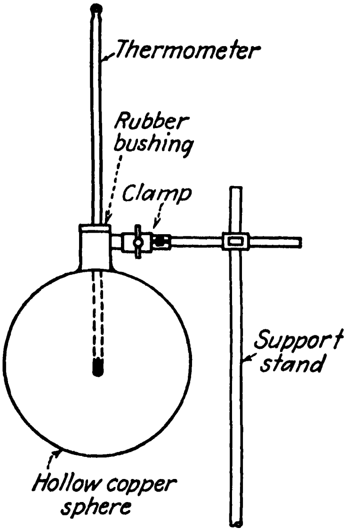

FIG. 35.—Globe thermometer.

At the present time the **globe thermometer** shown in Fig. 35 is perhaps the most practical instrument for measuring the radiant effect. It generally consists of a hollow 8-in.-diameter copper sphere, coated with black paint and containing an ordinary mercury thermometer with its bulb located at the center of the sphere.

Since the area projected by the globe is the same in all directions, the mean radiation value is not influenced by the direction from which radiation reaches the instrument. However, in considering a nonspherical body such as the human body, the direction of the source of radiation should be taken into account, since the projected area of the body in different directions may vary considerably. This point may assume some importance where the bulk of radiation is from a localized source, but its evaluation has thus far been difficult to measure and interpret.

When it is desired to measure the radiant effects from only a single localized area or surface, the radiation thermopile is considered a useful instrument.[1] This device consists of a number of blackened fine-wire thermocouples surrounded by a small cell and arranged so that only a small opening permits the radiation from an external source to strike the sensitive element.

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[1] J. D. HARDY, *Jour. Clin. Investigation*, Vol. 13, 593, 1934.

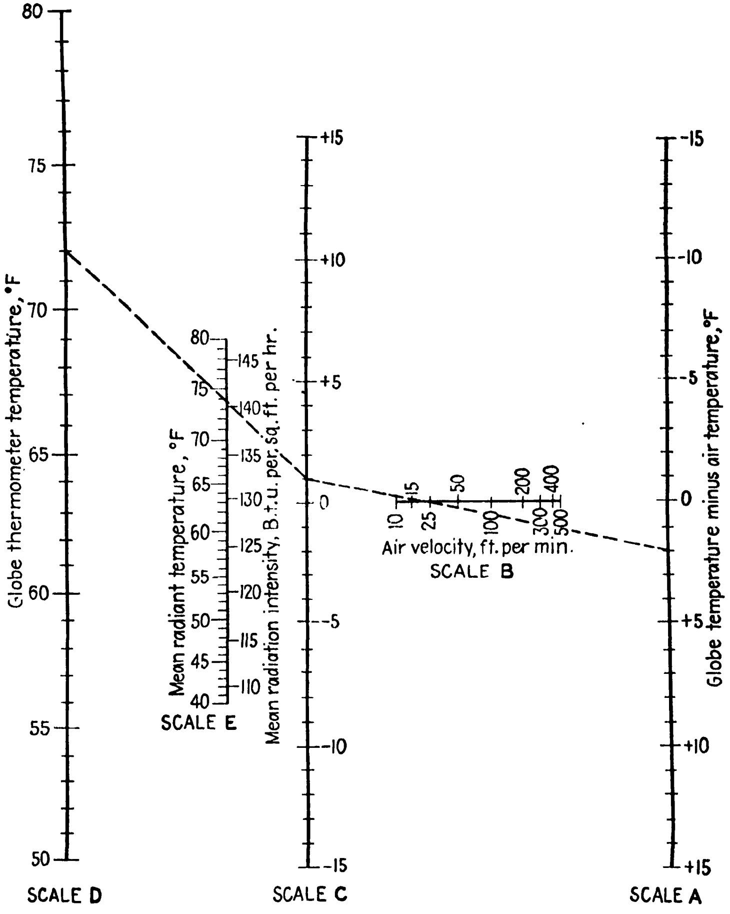

FIG. 36.—Chart for determining mean radiant temperature.

If the surfaces surrounding the globe thermometer are warmer than the air, the temperature recorded by the thermometer inside the globe will be above air temperature as measured by a separately suspended and shielded thermometer. Conversely, with the walls and other surroundings cooler than the air, the globe thermometer will be below air temperature. It has been found that the globe reaches equilibrium in about 15 min.

When the globe thermometer is in equilibrium with its environment, the effects of radiation and convection have been found to balance each other in accordance with equations derived by Bedford and Warner.[1] In establishing the convection loss from the instrument it is necessary to determine the air velocity over the globe, which in most rooms is less than 50 ft. per min. The anemometer referred to in Art. 319, Chap. XVII, is not capable of accurately measuring such low velocities, but the katathermometer has been found practical for this purpose. If a globe-thermometer reading, air temperature, and air velocity have been established for a given environment, it is possible to determine the mean radiant temperature from the alignment chart illustrated in Fig. 36.

Example.—Assume that a room heated by a direct cast-iron radiator has a globe thermometer suspended at the breathing level in the center of the enclosure and that it is desired to establish the mean radiant temperature of the environment. The globe thermometer shows a reading of 72°, the air temperature is 70°, and the air velocity as derived from a katathermometer is 25 ft. per min. Draw a line from scale A at plus 2°, which represents the difference between the globe and air temperature, through B at the 25-ft.-per-minute air-velocity point until it intersects C. From this intersection draw another line through the 72° globe-temperature value on scale D. The intersection on scale E gives 73.8° mean radiant temperature. Another value that may also be read from scale D is mean radiation intensity, which in this case is 140.5 B.t.u. per sq. ft. per hr. This latter value represents the mean intensity of all the heat-emitting and -receiving sources in the enclosure.

The significance of heat radiation to and from the human body, as discussed in Chap. XIV, is well appreciated from a physiological standpoint, but the design of a heating system is not yet sufficiently refined to take this factor intelligently into account. Instruments such as the globe thermometer have been found useful in evaluating the combined effects of radiant heat within an enclosure. However, much additional work is yet required to devise an instrument that will measure the combined effects of the thermal environment on human comfort.

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[1] BEDFORD and WARNER, "The Globe Thermometer in Studies of Heating and Ventilating," *Jour. Hyg.*, vol. 34, 1934. See also *A.S.H. & V.E. Research Rept.*, "Measurement of the Physical Properties of the Thermal Environment," *Heating, Piping and Air Conditioning*, June, 1942.

79. The Location of Radiators.—The location of the radiators in the room is extremely important. If they are placed along an inside wall, there is a tendency for uncomfortable drafts to be formed by the cooling effect of the windows and outer walls. The cold current of air thus formed flows without interruption across the floor. This "window chill" often causes extreme discomfort. It is best prevented by placing the **radiators directly beneath the windows,** especially in office buildings, schools, auditoriums, etc., where the occupants are stationary. High and wide radiators placed at the side of window openings do not produce as comfortable heating effects as long, low, and narrow radiators placed under windows.

80. Radiator Enclosures.—The cast-iron radiator is regarded as marring the appearance of some rooms, and various forms of cabinets and grilles are installed to conceal it; shields extending over the top of the radiator are also used in order to protect walls and hangings. These devices have a marked effect upon the heat emission and upon the heating effect of the radiator. Most of them reduce the total heat emission, because they restrict the convection currents or interfere with the radia-

TABLE 28—HEAT EMISSION OF ENCLOSED RADIATORS[1]

Type (Fig. 37)	Installation conditions	Heat emissions shall be altered by percentage indicated
1	When dimension *A* is as shown and dimension *B* is equal to 80 per cent of *A*.............	+10
2	When dimension *A* is as shown and dimension *B* is equal to 80 per cent of *A*..............	+ 5
3	When dimension *B* is equal to 80 per cent of *A*, dimension *C* is equal to 150 per cent of *A*, and dimension *D* is equal to *A*................	0
4	When dimension *E* is equal to 50 per cent of *A*.	−10
	When dimension *E* is equal to *A*...........	−20
	When dimension *E* is equal to 150 per cent of *A*......................................	−35
5	When dimension *E* is equal to *A*......	−30
6	When as shown..........................	− 5

---

[1] A.S.H. & V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings (Edition of 1929).

tion of heat or both; but in some cases it has been shown[1] that the heating effect from the standpoint of comfort and economy is as good as, or better than, that of a bare radiator, because of the tendency to project the currents of warm air outward into the occupied zone of the room rather than toward the ceiling. The essential features of a properly designed shield or enclosure are (1) the minimum possible restriction of the air flow, at the bottom as well as near the top; (2) a solid back to shield the wall from radiant heat; and (3) an outlet located as high up as practicable.

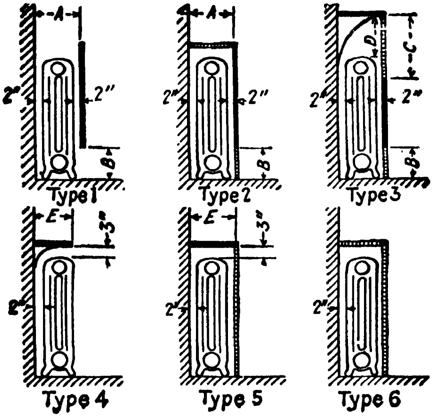

FIG. 37.—Various arrangements of radiator enclosures.[1]

It is possible to increase the heat emission by constructing an enclosure which by its chimney effect stimulates the flow of air over the radiator surfaces. Most enclosures, however, somewhat decrease the heat emission. Figure 37 shows several ordinary styles of enclosure, and Table 28 gives the effect on heat emission ordinarily allowed for in design.

81. Baseboard Radiators.—Another recently developed form of heating is a hollow cast-iron radiating element, designed to take the place of a conventional wood baseboard. The type shown in Fig. 38 is available in 1- and 2-ft. lengths, 7 in. high and 1¾ in. thick. The several sections are joined together by two push nipples and a short tie bolt. Any space not filled by the heating element is fitted with a piece of standard wood baseboard milled to resemble the cast-iron unit. Supply and return connections to each assembled unit are made as with conventional radiators. Baseboard panels will expand about ⅛ in. in 10 ft. for a temperature rise of 180°, and for that reason swing connections should be located in all branches. An accessible valve enclosure may be bolted to each baseboard panel at some inconspicuous place to control or regulate each unit individually. This particular design has a heat output of 261 B.t.u. per hr. per lin. ft. of section with water at 200° and 300 B.t.u. at 215°, the latter corresponding to the equivalent of 1.25 sq. ft. of conventional

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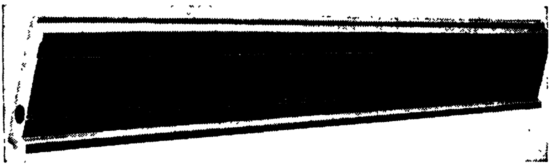
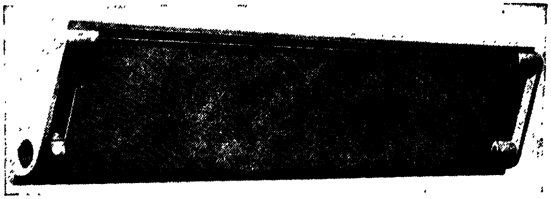
[1] *Univ. Ill. Eng. Exp. Sta. Bull.* 192.



(a) Front view.



(b) Back view.

Fig. 38.—Cast-iron baseboard radiator. (*Burnham Boiler Corporation.*)

steam radiation. These heat-emission values may be increased 60 per cent by using a variation of the above design, which consists of a hollow cast-iron baseboard section having fins cast integrally on the wall side of the unit. It is arranged so that it functions much like a convector with air openings on the top and bottom of each section.

The pure convection type of unit diagramed in Fig. 39 consists of finned copper pipe placed near the floor. It is hidden by a metal enclosure that resembles a baseboard. This heating element is of ¾-in. copper tubing with 2¼-in.-diameter fins spaced either three or five fins to the inch, depending on the desired output of the unit. Sections are available in standard lengths from 1 to 10 ft. The respective heat emissions per lineal foot of the three-fin and five-fin units, with water at 200°, are 440 and 550 B.t.u. per hr. In this system each floor of a structure or group of rooms is generally served by a continuous loop of piping extending along the outer walls. Where door openings exist, copper pipe without fins is dropped below each opening and brought back up to the baseboard level to provide for a continuous loop.

The amount of radiation required for either the cast-iron or finned copper units is determined by calculating the heat losses in the conventional manner and then using the proper heat-emission factor for obtaining the lineal feet of baseboard radiation needed. It is customary to place this type of radiation on outside walls, but an additional unit or units may be installed on inside walls where necessary.

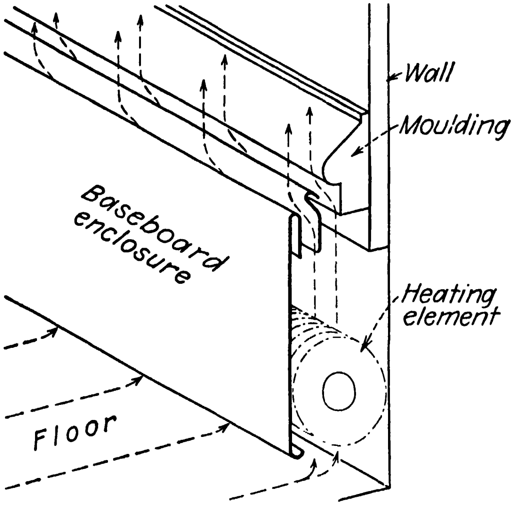


FIG. 39.—Copper convection-type baseboard unit. (*Warren Webster & Company.*)

Forced hot water is the heating medium customarily used on these systems, and therefore adequate vent lines and pet cocks should be provided for relieving the system of air when necessary. Advantages claimed for baseboard heating are (1) the heating element is practically invisible, (2) it permits wide latitude for interior decoration, (3) no wall or drapery discolorations, (4) low floor-to-ceiling temperature differentials, and (5) heat is released from a long, low panel and not from a concentrated area.

Recent tests[1] have indicated that with this type of heat distribution, floor-to-ceiling differentials of ¼° per ft. of height could be expected as compared with about 1° per ft. on conventional radiator and convector installations. Furthermore, with baseboard heating the temperature of the air at the 3-in. level above the floor was the same as that at the 30-in. level. Hence the tendency to produce a warm floor may be regarded as an important attribute of this type of heating. It is therefore particularly adaptable to basementless homes, where cold floors are prevalent.

PANEL HEATING

82. General.—Recent efforts to completely conceal heating equipment have resulted in an arrangement whereby the fluid, whether it be hot water, steam, air, or electricity, is circulated through distribution units embedded in the building construction. This form of heating has been widely used in England and other parts of Europe for many years, but not until recently has it had much acceptance in this country. Clarification is desirable to distinguish between the various types of distribution and heat-emitting surfaces employed in this type of heating.

The terms *panel heating* and *radiant heating* are often used interchangeably, but there are important differences in the two terms. They are properly defined as follows:

Panel heating is a method of introducing heat to rooms in which the emitting surfaces are usually completely concealed in the floor, walls, or ceiling. The heat is disseminated from such panels partly by radiation and partly by convection, the relative amounts depending on the panel location. Ceiling panels release the largest proportion of heat by radiation and floor panels the smallest. The proportion of heat disseminated by radiation and convection is also dependent to some extent upon panel-surface temperatures.

Radiant heating, in the pure sense of the term, does not refer to the type of surface but rather to the manner in which heat is disseminated from the surface. There are several forms of radiant heating, such as the common electric heater having a high temperature element and, usually, a reflector. A system using ceiling panels is also properly termed a radiant heating system. It is thus evident that the term radiant heating is applicable in varying degrees to several types of heating systems including, to a degree, the ordinary heating system equipped with cast-iron radiators.

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[1] Kratz and Harris, "A Study of Radiant Baseboard Heating in the IBR Research Home," *Univ. Ill. Eng. Exp. Sta., Bull.* 358, 1945.

The various types of panel heating systems may be classified according to the following outline:

- Panel heating
  - Hot water
    - Pipe coils embedded in floors, walls, or ceilings.
    - Circulated through recessed hollow metal panels or plates flush with interior surface finish (somewhat similar in form but generally larger than the baseboard radiation discussed in Art. 81).
  - Air
    - Gravity circulation through interior framework of building construction.
    - Mechanical or forced circulation through supply and return ducts to shallow plenum in ceiling or floor.
  - Electricity
    - Resistance wires embedded in wall coverings.
    - Wires embedded in floors or walls.

The basic advantage claimed for a panel heating system is that of comfort, because the convective heat loss from the human body is improved and the loss by radiation is reduced. The physiological aspects of this problem are covered in Chap. XIV. Actually the heat emission by radiation from the large heating panels raises the temperature of the unheated walls and furniture, and the result is a higher *mean radiant temperature* (see Art. 78).

Under the above conditions it is contended that lower air temperature may be maintained for equivalent comfort. European practice indicates that this comfort air temperature may be as low as 65°, but there is evidence accumulating in this country to indicate that the prevailing practice will set this temperature around 70°. It might be argued that this latter air temperature is comparable to design standards now being used with other forms of convection heating. It should also be realized, however, that in many cases the actual thermostat setting in this latter form of heating is in the range from 72 to 74° where radiation to cold walls and glass is particularly prevalent. Under such conditions the merits of panel heating become evident, even though the system seems to offer its greatest advantages in the form of an improved method of heat distribution by the attainment of greater uniformity of surface temperatures.

83. Theory of Heat Flow.—The principle of radiant heat transfer exists in such common examples as the outdoor sun and an open fireplace. None of the radiant heat emanating from these sources directly heats the air, but the enclosed surfaces and solid objects warmed by these rays release heat by convection, which warms the air (refer to Arts. 8 and 73). There are many instances where comfort is achieved with comparatively low air temperatures (50 to 60°), panel heating sources producing the necessary compensating radiant heat effect. Such examples are open-air schools and hospitals where a ceiling panel releases a major portion of its heat by radiation.

Heat transfer through a panel takes place by conduction, while the final transfer from the panel surface to the surroundings is by both radiation and convection. With the natural tendency of warm air to rise it becomes apparent, as indicated by the data in Table 29, that the convection component is greater with a floor panel than with a ceiling unit. Conventional heating systems have convection components ranging from 50 to 75 per cent, so that by comparison a panel heating system, irrespective of location, seems to produce the maximum now obtainable in radiant heat transfer.

TABLE 29.—TYPICAL PANEL PERFORMANCE DATA

Panel location	Maximum surface temp. range, °F.	Panel output, B.t.u. per hr. per sq. ft.	Per cent total panel output	
			Radiant transfer	Convection transfer
Ceiling	115–130	70–120	65	35
Wall	95–100	45– 80	60	40
Floor	80– 85	20– 50	55	45

84. Panel Location.—When it is realized that properly designed floor, wall, or ceiling panel systems may give equally satisfactory results, it becomes evident that the selection of panel location is largely a matter of personal judgment. Application of certain panels is frequently restricted by structural details specified by the architect. Other factors to be considered are type of occupancy, furniture or equipment location, large glass areas, heat-storing capacity of building construction, room height, possible change of wall partitions, climate, exposure, and first cost.

Ceiling panels are the most flexible and may be designed for the maximum radiation output, as indicated from the data in Table 29. The large percentage of direct radiation emitted from ceiling panels makes them preferable from the standpoint of restricting convection currents. However, this factor may be a disadvantage in rooms where heavy smoking occurs and natural ventilation is at a minimum. Ceiling heating has found wide application in multistory structures, where the proper location of the coils has also led to a satisfactory method of heating the room above by providing the proper amount of upward heat flow through the floor. With the hung-ceiling type of panel heating, the coils are covered only with a thin skin of plaster below the coils, and consequently this system is subject to shorter warming-up periods.

Wall panels have been used principally to counteract excessive wall losses and large glass areas. Also, where insufficient surface area exists in a ceiling or floor panel, it has been found convenient to locate the necessary additional coil surface in a wall.

Floor panels have found particular application in one-story buildings and basementless structures. In general, installation costs have been found lower for floor panels, and greater comfort is realized in the living or working areas with this type of heating. This design requires more exact requirements regarding limiting surface temperatures; no point on a floor should exceed 85°, as otherwise the discomfort of warm feet will result. With coils cast in heavy concrete slabs, more time is required for heating up. Moreover, the system is not particularly responsive to exact control, and overheating is apt to occur.

Where the pipes are buried in the structure, such as the ceiling, walls, or floor, there is a pronounced flywheel effect in the heating of the building, and therefore the system must be operated on the basis of continuous rather than intermittent heating.

85. Coil Design.—Various coil shapes may be used, but the two basic patterns are (1) the continuous or sinuous type of coil (Fig. 40*a*), and (2) the grid type (Fig. 40*b*) or a combination of the two (Fig. 40*c*). The *continuous* coil is less expensive to fabricate and more adaptable to designs where an irregular plan is involved.

Less over-all welding is generally required for a continuous coil than with the grid type. The limiting factor on the use of the continuous coil is the pressure drop through the unit, as certain economic factors need to be considered in balancing the first cost against operating cost of a high pumping head. To maintain a reasonable pressure drop through coils, prevailing practice has suggested the use of ½-in.-diameter pipe up to about 150 ft. in length, ¾-in. pipe up to 300 ft., and 1-in. pipe for longer lengths.

The *grid* type of coil may be designed to serve large areas where a low pressure drop is needed or more uniform surface temperature is desired. It is occasionally found desirable to concentrate the number of pipes within a single element to counteract excessive heat losses, such as are caused by large glass areas or unusual exposures. Where possible the heating medium should be supplied to the coil at the outside exposure, slowly cooling on its path through the coil and thus providing a desirable temperature gradient between the outside and inside walls.

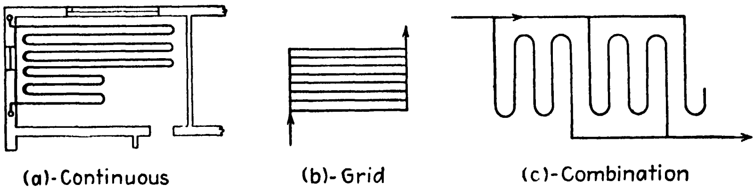
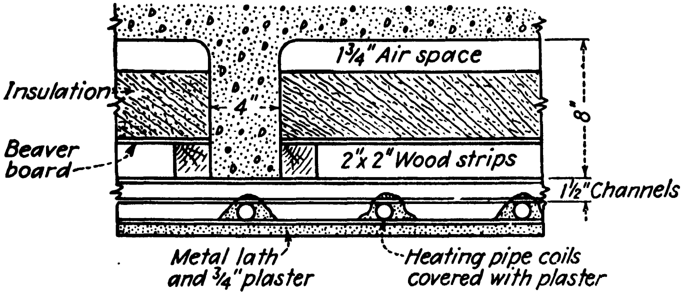

(a)-Continuous (b)-Grid (c)-Combination

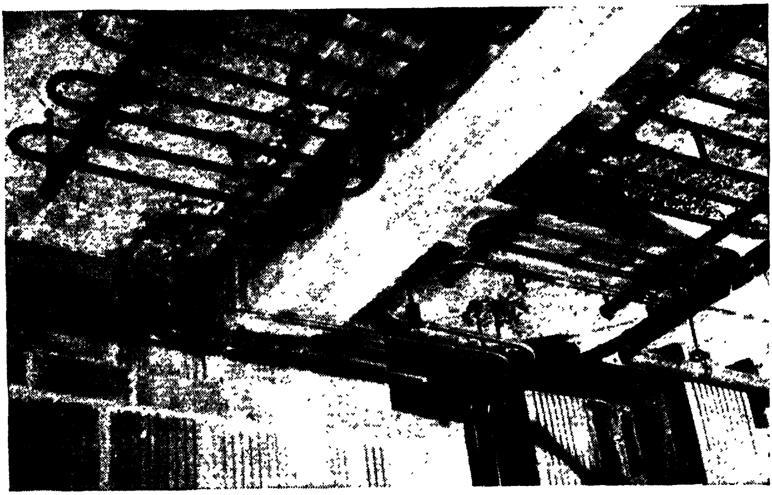
FIG. 40.—Typical coil-design shapes.

Materials.—Coils may be constructed of wrought-iron pipe, steel pipe, or hard or annealed copper tubing. Ferrous standard pipe sizes range from ½ to 2 in., and copper tubing from ¼ to 1 in. outside diameter.

Construction.—The large majority of *ceiling* panels have been installed with the coils suspended below the ceiling or floor construction, as shown in Fig. 41. Suitable channels or 1¼-in. standard steel pipes are supported below the floor construction on about 3-ft. centers and the coils are wired to these supports. It has been customary to provide adequate insulation above the coils and over the entire ceiling area to minimize the upward heat flow. Wire lath is placed directly underneath the coils, and a special plaster finish is applied. The first or scratch coat of plaster must be pushed through the wire lath so that it makes ample contact and practically surrounds the pipe. In some of these installations it is customary to work a material such as natural-color burlap into the finish or setting coat, for the purpose of eliminating plaster cracks. The plastering of such coils should not be attempted by anyone not familiar with the difficulties involved. The coils should be connected to the necessary supply and return piping and thoroughly tested before any plastering is done.



a. Cross section through ceiling construction.



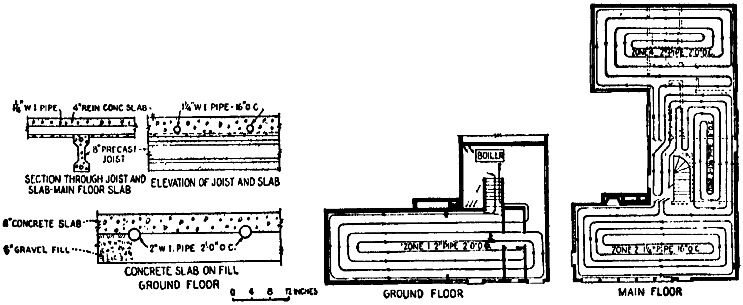
b. View of ceiling coil before plastering.

FIG. 41.—Ceiling-type heating panel. (*Wolff & Munier, Inc.*)

One arrangement of *floor* panel construction is diagramed and illustrated in Fig. 42. Views *a*, *b*, and *c* show progressive steps of the installation with the welding of a continuous-type coil shown in *a*, leveling the coil in *b*, and pouring the concrete slab in *c*. The location of coil in the slab construction is diagramed in *d*. It has been found that the amount of concrete fill over a pipe may vary from 1½ to 6 in. without materially affecting the coil performance.

a. Welding coil. *b*. Locating and leveling.

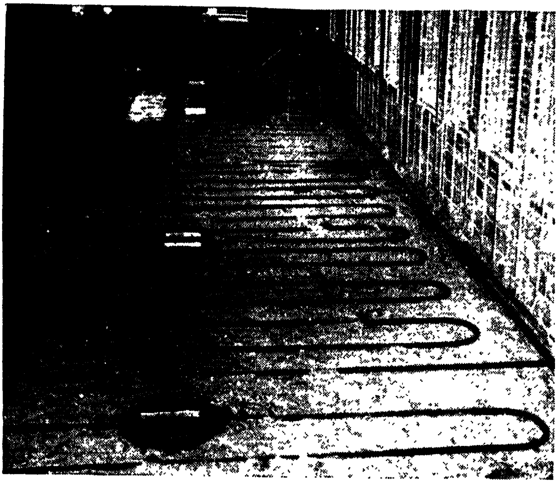
c. Pouring concrete slab.



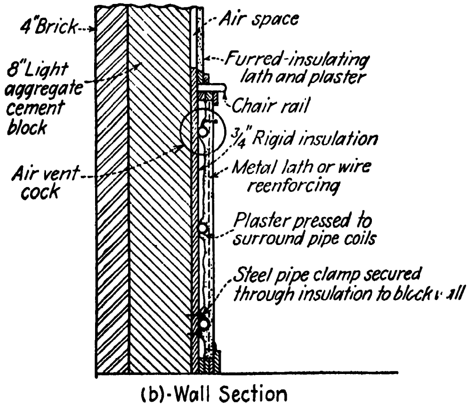
d. Section through slabs. *e*. Panel layout.

FIG. 42.—Floor type heating panel. (*A. M. Byers Co.*)

A combination *wall* and floor installation is shown in Fig. 43, together with a section through the wall construction. In this design each room has an individual pipe coil, which is arranged with a valve adjustment located in a sheet-metal floor box (see Fig. 44.)



a. View of floor coil and wall coil below window at rear.


4" Brick
8" Light aggregate cement block
Air vent cock
Air space
Furred-insulating lath and plaster
Chair rail
¾" Rigid insulation
Metal lath or wire reenforcing
Plaster pressed to surround pipe coils
Steel pipe clamp secured through insulation to block wall
(b)-Wall Section

b. Wall section.

FIG. 43.—Combination wall and floor panels. (*Reprinted by permission of Heating and Ventilating.*)

86. Air Systems.—Records dating back to the ancient Romans have disclosed panel heating systems constructed of masonry ducts built within side walls or beneath floors, through which hot flue gases were made to flow.

In recent years this method of circulating warm air through shallow ducts has been effectively applied to a few cathedrals and large public buildings, but in such buildings the design is more expensive to construct than embedded pipes, and for that reason has not been extensively used.

A few residences have utilized this principle both with gravity and forced[1] warm-air circulation. Even though an indirect hot-water or steam coil is used for warming the air, it is necessary to provide fireproof construction for the spaces through which the air is circulated. This may be done by first applying a sheet-metal panel to the bottom of the ceiling joists. Under this panel and also to the joists are attached 2½-in. metal strips, from which the metal lath and plaster are suspended to form the necessary shallow air space.

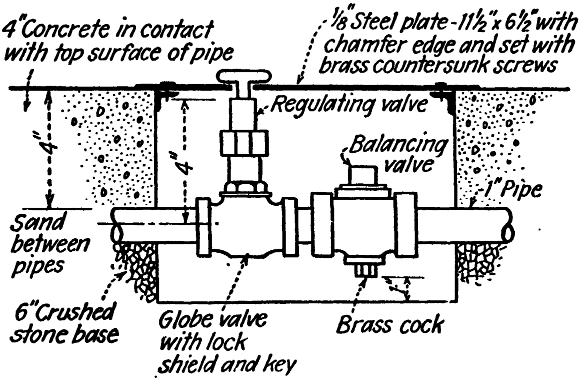

4" Concrete in contact with top surface of pipe
1/8" Steel plate - 11½" x 6½" with chamfer edge and set with brass countersunk screws
Regulating valve
Balancing valve
4"
4"
1" Pipe
Sand between pipes
6" Crushed stone base
Globe valve with lock shield and key
Brass cock

FIG. 44.—Detail of valve adjustment for each room coil. (*Reprinted by permission of Heating and Ventilating.*)

87. Design Procedure.—Extensive calculation procedures for designing panel heating systems, involving the computation of the mean radiant temperature,. have been developed in England[2] and this country.[3] The authors recommend, however, that the conventional B.t.u. method based on the heat losses from the building be utilized, at least until more experience is gained with the new methods.

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[1] RANDOLPH and WALLACE, "Performance of a Residential Panel Heating System," *Trans. A.S.H. & V.E.*, 1943.

[2] HENRY R. BRUCE, "Panel Warming Calculations," *Jour. Inst. Heating & Ventilating Engrs. (London)*, December, 1936

[3] Note references at end of chapter.

The use of the B.t.u. design procedure is particularly adaptable in this country, as the mean radiant temperature of the environment is likely to be only 1 or 2° above the air temperature, and under such conditions the design resembles a conventional radiator or convection system insofar as heat-loss calculations are concerned.

Panel Area.—To determine the panel area it is desirable first to compute the heat loss of the room or building according to the procedure outlined in Chap. II. Next divide the heat loss by the heat output per sq. ft. of panel. This latter value is dependent on the panel location, panel-surface temperature, coil design, and pipe size and spacing.

The surface temperatures listed in Table 29 are considered typical, but exact research data are lacking to indicate the heat transfer of embedded pipe in various types of construction and with different types of floor or ceiling coverings. The heat-output data in Table 30 may be considered fairly reliable for ceiling panels.

Experience has shown that a value of 3.5 B.t.u. per hr. per sq. ft. of external pipe surface per degree Fahrenheit temperature difference between the water and air is conservative for estimating the heat output of pipe surrounded by solid masonry. Even though definite data are lacking, a comparable value for pipes located between joist in an air space with double flooring is around 2.0 B.t.u. per sq. ft.

TABLE 30.—HEAT OUTPUT OF CEILING PANELS
(Pipes buried in plaster, similar to Fig. 41a.)

Temp. diff., °F., pipe water to air	Heat output, B.t.u. per hr. per sq. ft. of panel surface				
	Spacing ½-in. diameter iron pipe, in.				
	4½	6	9	12	18
20	28	24	17	13	9
30	46	40	28	22	15
40	69	60	42	32	23
50	92	80	56	43	30
60	115	100	70	54	38
70	142	123	86	66	47

Example.—A large office space 20 ft. wide by 50 ft. long by 10 ft. high and having a heat loss of 40,500 B.t.u. per hr. is to be provided with an invisible panel-heating system designed to maintain a uniform air temperature of 70°F. with a maximum water temperature circulated through the coils not to exceed 140°F. (1) Determine the spacing of ½-in.-diam. pipe and the total length of sinuous coils required for a ceiling installation. These coils are to be arranged in twelve 4 ft. by 10 ft. decorative panels similar to the design illustrated in Fig. 41. (2) For comparison, also design a floor-panel system comparable to the arrangement shown in Fig. 42*d*, with the same size pipe used in (1) and determine the pipe spacing and length of coil required.

1. With insulation above the coils, it can be assumed that 90 per cent of the heat flow will be downward and 10 per cent into the space above. Therefore, the ceiling panels must be designed to emit 40,500/0.90 = 45,000 B.t.u. per hr.

Assuming a 20° temperature drop through the coil, the resulting mean water temperature is (140 − 10), or 130°. With a room-air temperature of 70°, the temperature difference is 60°. The panel area available for coils is 12(4 × 10) = 480 sq. ft., and the resulting heat-emission rate is therefore 45,000/480 = 94 B.t.u. per hr. per sq. ft. surface.

From Table 30 it can next be determined that the above requirements will be met with a ½-in.-diam. pipe spaced 6 in. on centers. The length of coil required can be estimated as follows: Across each panel there would be 8 pipes spaced 6 in. on center, with each pipe 10 ft long or a total of 80 ft. per panel. With 12 panels the total length would be 80 × 12 = 960 lineal ft.

2. It can also be assumed that there will be a downward heat loss from the floor panel of 10 per cent, and it is therefore necessary to design the system for a total load of 45,000 B.t.u. per hr.

Using the heat transfer value of 3.5 B.t.u. per hr. per sq. ft. of external pipe surface per degree Fahrenheit temperature difference between water and air for pipes embedded in concrete, it is possible to calculate the pipe spacing required. Assume a floor-panel output of 45 B.t.u. per sq. ft. from Table 29, which is on the conservative side. For the same temperature difference (60°) used in (1), the square foot of external pipe surface required to produce the necessary heat emission would be 45 divided by 3.5 B.t.u. × 60° or = 0.215 sq. ft.

By reference to Table 45 (Chap. VIII) it will be noted that this corresponds closely to the ½-in.-diam. pipe desired, spaced 12 in. on centers. The same approximate heat-emission rate could also be obtained with a 1¼-in.-diam. pipe spaced 24 in. on centers, as it has about twice the external surface, or 0.435 sq. ft.

If the ½-in.-diam. pipe was used, there would be 19 pipes spaced 12 in. on centers across the room and about 49 ft. long or a total of 930 lineal ft.

Coil Layout.—After the coil length and spacing have been established, it is desirable to arrange the layout so that the coils are distributed over the entire panel area. The coils should be installed as near level as possible with means provided for proper drainage and the necessary venting. The circulating pump, expansion tank, and necessary supply and return piping to each coil may be selected on the same basis as for any other forced hot-water heating system (Chap. X).

Control.—It is customary to arrange the piping layout so that the coils may be individually controlled, balanced, and zoned from some accessible location, as indicated in Fig. 42*e*. The most desirable automatic control for a panel heating system is the arrangement outlined in Fig. 154, where the water temperature is adjusted in accordance with the outdoor temperature. To prevent overheating, a room thermostat should be provided, either to stop the circulating pump or preferably to lower the coil water temperature.

Summary.—It is probable that the acceptance of true invisible panel heating will largely depend on whether the application cost can be reduced to make it comparable to other forms of heat distribution now available. Some of the advantages and disadvantages of panel heating are summarized herewith.

Advantages:

a. Eliminates all visible radiators and grilles, thus saving valuable floor or wall space.

b. Full freedom for interior decorative treatment, location of furniture, and placement of machinery or office equipment in the case of commercial establishments.

c. Use of ceiling or floor panels permits rearrangement of interior partitions.

d. Reduces surface discoloration and high temperature concentrations, which often cause streaking and dust deposits.

e. Reduces convective drafts and stratification, thus avoiding zones of discomfort, particularly at floor level. Floor-to-ceiling temperature gradients of about ¼° per ft. of height are obtainable.

f. Since the installation is completely enclosed, tampering with fixtures is impossible, which is an advantage in schools, hospitals, offices, and industrial plants.

Disadvantages:

a. Cost of installation apt to be more expensive than conventional heating systems, especially if precautions are taken to ensure against plaster cracking.

b. Requires close cooperation of architect and builder in designing and installing the system.

c. Unless the system is carefully zoned and balanced, hot surface spots that are difficult to correct are apt to develop.

d. The heat capacity of some heavy masonry panels may cause overheating and slow warming up.

e. Difficult to make changes, additions, or repairs to an existing installation.

UNIT HEATERS AND VENTILATORS

88. Definitions.—Unit heaters are convection heaters in which the circulation is produced by a fan. The heating surface and fan are incorporated in a factory-built unit. The fan creates sufficient velocity to discharge the heated air through a considerable distance, and this effect is utilized to obtain proper distribution in a large room.

There are several types of units, which are defined as follows:

Unit heaters are units consisting of a fan and heater enclosed in a casing. They are intended only to recirculate and warm the air in a room.

Unit ventilators draw all or part of their air supply from outside and usually include a filter and sometimes a humidifier.

Unit conditioners are units that cool and dehumidify. Sometimes they are also designed to heat and ventilate. These will be discussed in a later chapter.

89. Unit Heaters.—Unit heaters are classified as (*a*) suspended type, (*b*) floor type. The suspended type is the more common. It is usually installed overhead in the upper part of the space to be heated, whereas the floor type is mounted directly on the floor and discharges air at a lower elevation than the suspended type.

Steam is usually used as the heating medium, although hot water can be used if desired. In general, the heating elements are nonferrous and have extended surfaces, though in most electric heaters a grid or strip type of heating element is used. The direct-fired gas unit heater has the burners installed within the heat-transfer section.

The principal advantages claimed for unit heaters are

a. Large capacity; therefore a small number of units is required.

b. Piping system simplified.

c. Cost of heating system reduced.

d. Space saved, because heaters are compact and can be located where they will be out of the way.

e. Rapid heating.

f. Heat directed where wanted.

g. Wide distribution of heat.

h. Difference between floor and ceiling temperatures reduced.
i. Economical operation.
j. Circulation of air obtained.
k. Ventilation made possible by installing fresh-air connection.
l. Heat output easily controlled.

The most common form of suspended unit heater is that shown in Fig. 45, in which a disk or propeller fan is mounted in the casing directly behind the heating surface. The fan takes in air from behind the heater and blows it over the heating surface and out into the room. The direction of the air stream is controlled by louvers on the front of the heater. Some heaters are of the pull-through type; that is, the air passes through the heating surface before going through the fan. In order to obtain greater capacity in a single unit, some heaters are built with two fans side by side behind a heating surface of double width.

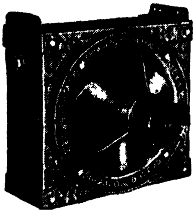

FIG. 45.—Unit heater of the suspended type. (*Buffalo Forge Company.*)

The suspended type of heater is well adapted to industrial use because it can be located out of the way above the heads of the workers. It can be suspended from roof trusses or columns or from wherever convenient. The smaller sizes are light in weight and in some cases are suspended from the piping. These heaters are easy to install and make possible a simple piping layout. Steam and return mains can be run overhead, where they can easily be given ample pitch and where they are out of the way. This eliminates troublesome piping difficulties with doorways and obstructions and makes underground piping unnecessary.

One form of suspended-type unit heater that is sometimes quite convienent to install is the overhead design shown in Fig. 46. The heating surface is horizontal with the fan above. The air enters at the top and is discharged through directional nozzles.

Some heaters are equipped with recirculating ducts extending to the floor, as in Fig. 47. This arrangement is used so that cold air can be pulled off the floor and the floor temperature increased. Heaters without such recirculating ducts take their supply of air from the level at which they are located and thus may tend to restrict the circulation of air at the floor level. The louvers on the front of the heater should be adjusted to direct the stream of heated air slightly downward so that the heat will be delivered where it is wanted and so that the heated air can mix with the cooler air at the floor.

For use in stores, auditoriums, and other places where appearance is an important factor, heaters are frequently concealed in the wall, in columns, or in an adjoining room. The recirculating duct is built into the wall with the heater, and the only parts of the installation that need to be visible are the grille at the floor and the grille in front of the heater. Figure 48 shows such an arrangement.

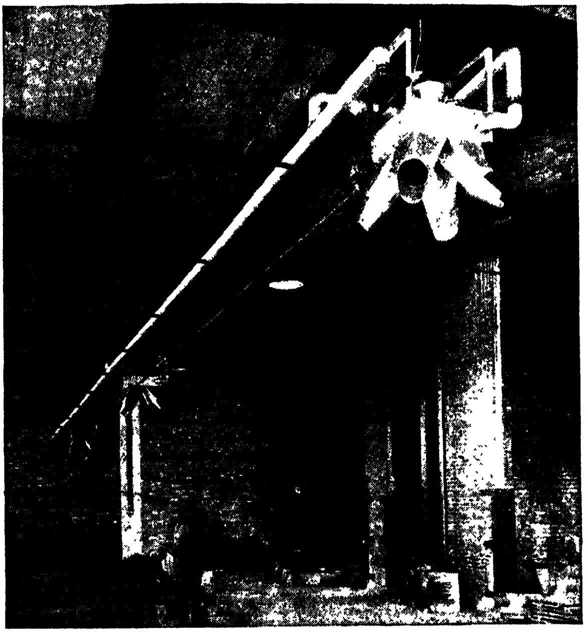

FIG 46.—Unit heater of the overhead type. (*L. J. Wing Manufacturing Company.*)

90. Floor Type.—This type of unit heater, illustrated in Fig. 49, generally consists of a metal housing within which are located the heating elements and one to four multiblade fans. The fans are mounted on a single shaft, which is driven by a motor located outside the housing. The heating elements are in the fan suction chamber so that the air, after entering the bottom of the heater,

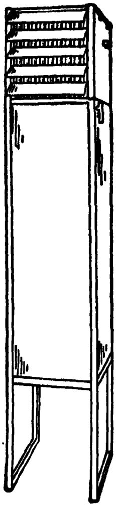

FIG. 47.—Unit heater with recirculating connection.

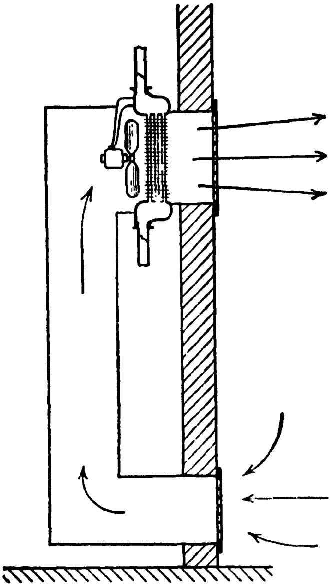

FIG. 48.—Unit heater installed in concealed location.

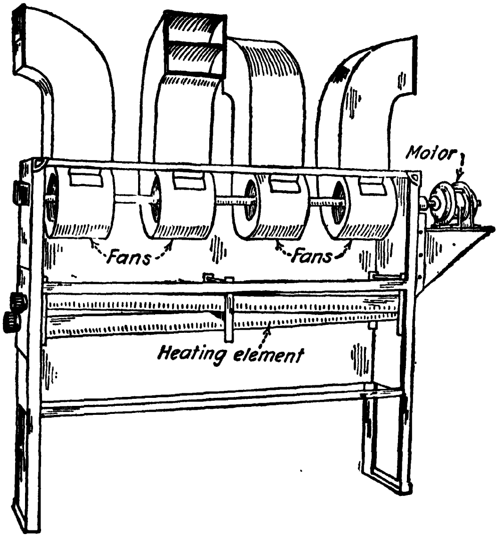

FIG. 49.—Interior view of floor-type unit heater.

is drawn over the heating surface before entering the fans. Each fan discharges through a separate short hooded duct in the top of the heater. The hooded outlets discharge the heated air horizontally at high velocity, just above the heads of the workers. The hoods can be arranged for discharging in different directions.

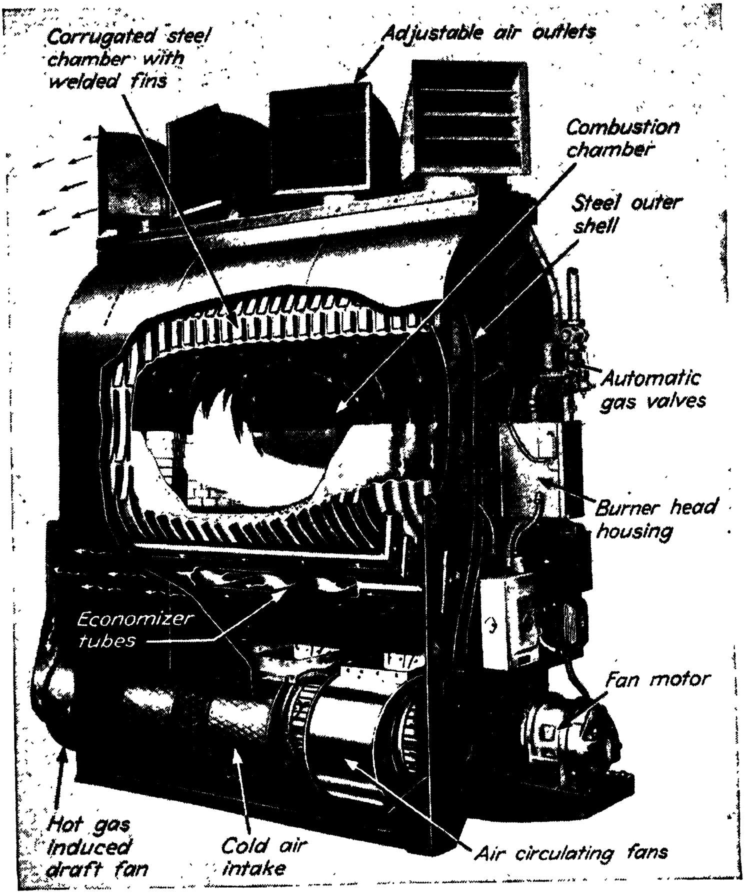

FIG. 50.—Gas-fired self-contained industrial heater. (*Dravo Corporation.*)

These heaters have greater capacity than the ordinary suspended-type unit heater, an output up to 1,000,000 B.t.u. per hr. being possible from a single unit. This makes it possible to heat large spaces with a small number of heaters. Another advantage of this type is that the air supply is always taken from the floor, thus tending to decrease the temperature difference between floor and ceiling. Where floor space is not available, this type of heater in somewhat modified form can be suspended overhead and used as a suspended-type heater.

The centrifugal fans used in the heater shown in Fig. 49 are inherently capable of developing a higher pressure than the propeller-type fan used in most suspended-type heaters. They can therefore discharge the air at a higher velocity and through greater distances.

Self-contained direct-fired floor-type heaters utilizing either automatic coal, oil, or gas fuel-burning equipment have been developed in capacities ranging from 400,000 to 8,000,000 B.t.u. per hr. These heaters have been adapted to heating large industrial spaces and airplane hangers. A gas-fired unit is shown in Fig. 50.

91. Air Temperature and Velocity.—The temperature of the air leaving a given unit heater or ventilator is determined by the entering-air temperature, the air velocity, and the steam pressure. Increasing the velocity decreases the final temperature; increasing the steam pressure or the entering-air temperature increases the final temperature. The heat output decreases as the entering-air temperature increases.

A given heat output may be obtained by discharging a small volume of air at high temperature or a large volume of air at low temperature. Heaters discharging low-temperature air at high velocity produce more uniform temperatures from floor to ceiling. This is because the lower temperature air does not have so great a tendency to rise and because the high velocity brings about a wide distribution and more thorough mixing and cooling of the warm air before it has a chance to rise to the ceiling. A fuel saving results, since a smaller quantity of heat is required to produce the desired temperature in the working zone. The large floor-type heaters are of the high-velocity type. The stream of air discharged from a unit heater induces a circulation of the air near it and in that way brings about a mixing of heated air and cooler air in the rooms. High-velocity discharge increases the induced circulation.

Unit heaters, such as those of the floor type, having a high outlet velocity, can be spaced much farther apart than the lower velocity heaters. One rule for spacing unit heaters is that air at a temperautre of about 135° can be spread horizontally about 10 to 15 ft. for every 100 ft.-per-min. outlet velocity. Increasing the delivery temperature decreases the horizontal distance that the stream of air will travel. Floor-type unit heaters having an outlet velocity of 1,500 ft. per min. have been successful when placed as far as 400 ft. from each other, but in general they should be spaced not more than 200 ft. apart. Suspended-type heaters having outlet velocities of 300 to 600 ft. per min. should be spaced from 50 to 100 ft. apart.

Unit heaters should be placed so that they assist each other in causing a circulation of air in the room, rather than discharging in a haphazard way and setting up cross currents. A much better distribution of heated air and a better circulation will be obtained if the heaters are arranged to discharge in an orderly fashion. One method is to arrange the heaters near the wall, as in Fig. 51, so that they will discharge tangentially around the room, thus causing a rotary circulation of air. Another scheme is to place them in the middle of the room and have them discharge toward the outside walls. The first scheme is to be preferred.

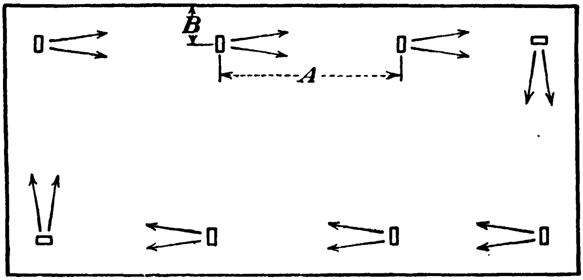

FIG. 51.—Arrangement of unit heaters. Distance *A* should be 50 to 100 ft.; distance *B* should be about 6 ft.

92. Temperature Control.—The usual method of control is to start and stop the fan manually or by a thermostat. Many fans are equipped with variable-speed motors, manually controlled, thus making it possible to adjust the speed to the desired heat output and keep the fan operating continuously. When the fan is stopped for short intervals, steam is usually not turned off. The heat output during times when the fan is not running is approximately 5 per cent of the capacity of the heater.

For fans controlled by thermostats, a control can be added to the thermostat circuit, which will prevent the fan from operating unless steam is turned on. This control may be a pressurestat, which will not permit the fan to start unless steam pressure is available, or it may be a thermostat connected to the return line between the heater and the trap, shown in Fig. 52, which will prevent fan operation unless the water is sufficiently warm.

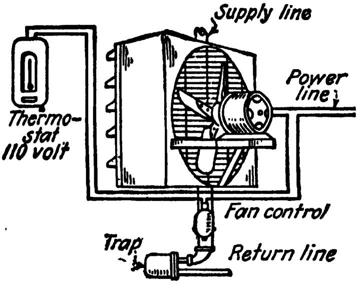

FIG. 52.—Unit heater controlled by room thermostat and by thermostat in return line.

Another method of control is to have one thermostat operate the steam valve and another in the return line control the fan. With this form of control, steam is turned on only when the room thermostat calls for heat, and the fan operates only when there is steam in the heater. One room thermostat can be used for controlling a battery of heaters if desired.

93. Piping Connections.—Because of the large capacity of unit heaters, care should be taken to see that the steam piping leading to them is of sufficient size. Unit heaters should not be used on one-pipe systems. If the heating system contains direct radiators operated with steam under vacuum, it is best to have the unit heaters served by a separate main, so that steam above atmospheric pressure can be supplied to the unit heaters, if desired, without interfering with the operation of the direct radiators.

The trap in the return line must have ample capacity to care for the large amount of condensation when the air supply to the heater is at a low temperature. A float trap may be used for this purpose or preferably a combination float and thermostatic trap (described in Chap. VIII).

94. Unit-heater Ratings.—A standard method for testing and rating unit heaters has been adopted by the A.S.H. & V.E.[1] Most manufacturers give the rating of their product as determined by this standard method.

95. Unit Ventilators.—One of the principal applications of unit ventilators has been in school ventilation. The type of unit developed for that purpose has been applied to other buildings. The schoolroom type of unit ventilator is low in height for locations beneath windows, and is often equipped with a filter in addition to the heating element and fan. An outdoor connection is

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[1] A.S.H. & V.E. Standard Code for Testing and Rating Steam Unit Heaters, *Trans. A.S.H. & V.E.*, 1930.

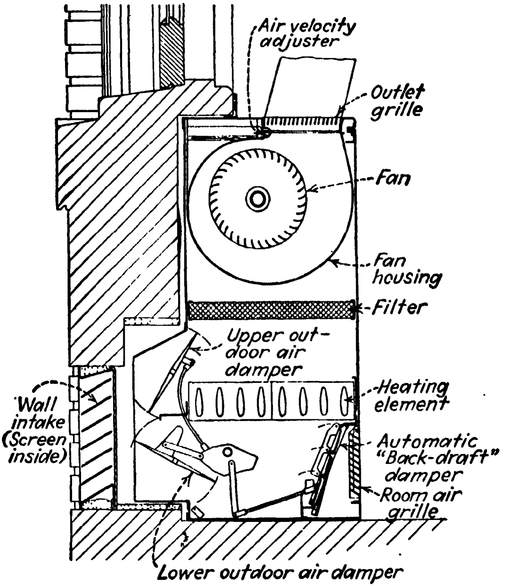

FIG. 53.—Unit ventilator (cross section). (*Herman Nelson Corporation.*)

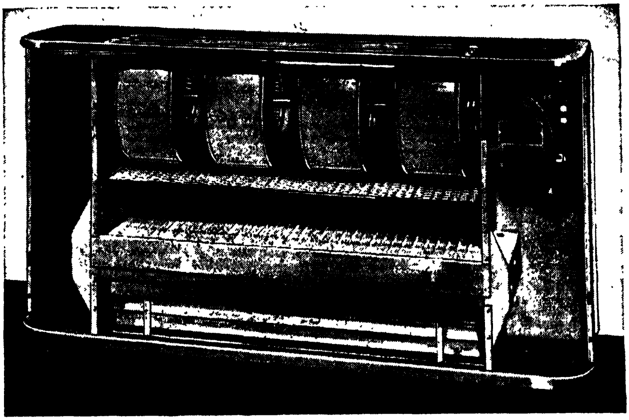

FIG. 54.—Unit heater of Fig. 53 (casing removed). (*Herman Nelson Corporation.*)

made for a fresh-air supply, and there is also provision for recirculation.

One design is shown in cross section in Fig. 53 and with front removed in Fig. 54. The essential elements are the extended surface heating element, the fan, and the filter. Air may be taken from outside or inside or both.

The unit ventilator for schoolwork was developed to take the place of the central fan system, which is expensive and which for various reasons had come into disrepute. The ventilation of schools has always been somewhat unsatisfactory, partly because of its inherent difficulties but also because of improper design and lack of intelligent operation. The unit ventilator has become popular because of its lower cost and also because it places the control of the system more in the hands of the teacher. There is still, however, much room for improvement in school ventilation.

In schoolrooms the unit ventilator is often supplemented by one or more radiators to take care of part of the heating requirements. Temperature control is usually in the hands of the teacher, who is provided with a control handle that operates the radiator valve, unit-heater steam valve, and recirculating- and fresh-air dampers.

96. Rating.—The revised Standard Code for Testing and Rating Steam Unit Ventilators was adopted in January, 1932, by the A.S.H. & V.E. It provides that the standard basis of rating is to be dry saturated steam at 2 lb. gage pressure (at 29.92 in. barometric pressure) at the heater coil, and air at 0° entering the heater when the heater is free of external resistance. Air volume is to be converted to standard air at 70°. Rating tables are to give, for various inlet and outlet temperatures, the heating capacity in equivalent direct radiator surface (240 B.t.u. per sq. ft. per hr.).

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Problems

1. Given the following heat losses from the several rooms in a building, choose suitable radiators. Room temperature 70°, steam temperature 215°. (Any one of several combinations of height and width will answer, although, in an actual building, space limitations would have to be considered.)

Room number	Heat loss, B.t.u. per hour	Remarks
1	11,300	Use tube-type radiator.
2	7,160	Use wall radiator.
3	23,900	Use two radiators 14 in. high—tube type.
4	21,400	Use tube-type radiator.
5	10,760	Use pipe coil placed vertically, pipes horizontally.
6	13,850	Use two wall radiators.
7	19,280	Use tube-type radiator.

2. Using the Stefan-Boltzmann and the Rice-Heilman equations of Art. 73, determine what percentage of the total heat from a cylinder is emitted by radiation (*a*) when the surface is highly polished copper and (*b*) when it is of rough steel. Assume diameter of cylinder, 12 in.; surface temperature, 215°; temperature of air and surrounding objects, 70°.

3. For the same rooms as in Prob. 1, and using the same kinds of radiators, make the selections assuming a room temperature of 60° and a steam pressure of 6 lb. gage.

4. A test of a 64-sq. ft. radiator gives 41.2 lb. of water collected in 2 hr. 20 min. The steam pressure is 3 lb. per sq. in. gage, and the steam quality is 97 per cent. The room temperature is 67°. What would be the heat output under standard conditions?

5. A hot-water convector test shows a B.t.u. output of 4827 with water entering at 197°F. and leaving at 177°. Temperature of air entering, 69°. What would be its heat output at an entering-water temperature of 190° and standard air temperature?

6. If a radiator emits 18,760 B.t.u. with steam at 2 lb. gage and room air at 68°, what will be its heat output with steam at 7 in. Hg below atmosphere and a room temperature of 72°?

7. A building contains 2,780 sq. ft. of steam-radiator surface, which is just adequate with steam at 215° to maintain 70° in the rooms. The system is to be converted to a hot-water system with a supply temperature of 170°. How much radiator surface must be added?

CHAPTER VI

FUELS AND BOILERS

97. Coal.—Before taking up the subject of boilers, it is desirable to study the various kinds of fuel and the general principles of combustion. Coal, coke, oil, and gas are used as boiler fuels.

The chief constituents of coal are carbon, hydrogen, oxygen, and nitrogen. The carbon exists partly in an uncombined state and partly in combination with the hydrogen and oxygen as hydrocarbon compounds that are given off as gases when the coal is heated. Coals are classified as anthracite, bituminous, etc., according to the relative proportions of fixed carbon and volatile matter, as given in Table 31.

All coals contain more or less noncombustible matter, consisting principally of moisture and ash. The nitrogen in the coal is also a noncombustible, but it is customary to treat it as combustible matter. The moisture content of different coals varies from 2 per cent to as much as 20 per cent, and the ash content from 4 to 20 per cent by weight of the coal as mined. It will be noted that the percentages in Table 31 are based on 1 lb. of coal as received. The moisture content varies depending upon the weather during transit, but the figures may be considered as average.

TABLE 31.—ANALYSES OF TYPICAL COALS
(On basis of composition as received)

Type of coal	Moisture, per cent	Ash, per cent	Volatile matter, per cent	Fixed carbon, per cent	Sulphur, per cent	Heating value, B.t.u. per lb.	Ash-softening temp., °F.
Anthracite..........	4.2	9.7	4.0	82.1	0.6	12,800	2600–3000
Semianthracite.....	2.3	10.0	10.0	77.7	0.6	13,500	2200–3000
Semibituminous....	3.5	3.0	21.0	72.5	0.7	14,600	2400–2600
Bituminous........	3.0	7.0	33.0	57.0	0.8	13,500	2200–2900
Subbituminous.....	20.0	7.9	34.3	37.8	2.0	9,500	1900–2600
Lignite............	38.4	6.6	25.8	29.2	0.8	6,800	2000–2500

Anthracite or hard coal is principally used for domestic and other purposes where a smokeless coal is required. It contains a high percentage of fixed carbon, is noncoking, and neither swells nor softens when burning. This coal ignites slowly but burns steadily with a short blue flame. It has a high density around 52 lb. per cu. ft. and does not crumble easily. The commercial sizes are listed in Table 32.

TABLE 32.—COMMERCIAL SIZES OF ANTHRACITE COAL

Kind of coal	Will pass through	Will not pass through
Barley	3/16-in. mesh	3/32-in. mesh
Rice	5/16-in. mesh	3/16-in. mesh
Buckwheat	9/16-in. mesh	5/16-in. mesh
Pea	13/16-in. mesh	9/16-in. mesh
Nut	1 5/8-in. mesh	13/16-in. mesh
Stove	2 7/16-in. mesh	1 5/8-in. mesh
Egg	3 1/4-in. mesh	2 7/16-in. mesh
Broken	4 3/8-in. mesh	3 1/4-in. mesh

Semibituminous or low-volatile coal ranges in volatile content from 14 to 23 per cent and is comparatively low in ash and moisture content. Due to its low volatile content, it can be burned with less smoke and is therefore sometimes called "smokeless" coal. Because of its soft structure the percentage of fines is increased materially through handling. These coals are commonly referred to as Pocahontas, which is the name of one of the most important seams in this group.

TABLE 33.—COMMERCIAL SIZES OF BITUMINOUS COAL

Kind of coal	Will pass through bars spaced	Will not pass through bars spaced
Lump	....	1¼ inches
Nut	1¼ inches	¾ inch
Slack	¾ inch	

Bituminous or medium-volatile coal is generally low in ash and moisture content, with from 23 to 33 per cent volatile. Its burning characteristics are similar to the low-volatile group, yet it is somewhat flashier and will ignite and burn more rapidly because of its volatile content.

High-volatile bituminous coals, mined extensively in many sections of United States, range in heating value from 11,000 to over 14,000 B.t.u. per lb. The moisture content ranges from 2 to 12 per cent. The ash and sulphur content, as well as the ash-fusion temperature, will vary widely. Some of these coals are extremely hard coking or "caking" and others are free burning or "noncaking." The structure is usually harder than the semi-bituminous group, but not so hard as anthracite.

Subbituminous coal is comparatively high in moisture and tends to air-slake or break up when exposed to weather. It is usually free burning, and the heating value ranges from 8300 to 11,000 B.t.u. per lb. All bituminous coals burn with a characteristic yellow flame, have an average density of about 48 lb. per cu. ft., and are sold in sizes corresponding to those outlined in Table 33. Slack coal does not command so high a price as the larger sizes because of its higher ash content and the difficulty of burning it.

Lignite contains a very high percentage of moisture and has a low heating value and a greater tendency than subbituminous to air-slake as it dries. It produces small quantities of smoke or soot, but because of the high moisture and ash content, the maximum burning rate is low.

98. Composition and Analysis of Coal.—Coal consists of carbon, hydrogen, sulphur, oxygen, and nitrogen combined in various ways, together with moisture and ash. The moisture includes both that originally contained in the coal and that acquired during storage and shipment. The moisture content of a given coal is determined by subjecting a finely powdered sample to a temperature of about 220°F. for about 1 hr. and noting the loss in weight during that time. This method, although not giving an absolutely accurate result, is the one universally employed.

The amount of volatile matter is determined by subjecting a sample of dried coal to about 1750° in a closed crucible until there is no further loss of weight and noting the decrease in weight. The residue left after distilling off the volatile matter consists of the fixed carbon and ash. By burning the sample in an uncovered crucible the fixed carbon can be removed, leaving the ash.

There are two forms of **coal analysis**—the "proximate analysis" and the "ultimate analysis." The former consists of a determination of the moisture, volatile matter, fixed carbon, and ash, in the manner just described. This is the more useful form of analysis and is the one generally used by engineers, as it serves to show the type of coal and its more important characteristics. Although determined separately, the sulphur content is frequently reported with a proximate analysis in order to give a better indication of the characteristics of the ash. The ultimate analysis, which consists of a determination of the carbon, hydrogen, oxygen, nitrogen, and sulphur, is necessary only when a close study of the combustion of coal is being made. In the proximate analysis, the percentages may be reckoned on a basis of either dry coal or coal "as recieved." In the former case the moisture content is given in addition.

The heat value, or **calorific value,** of a fuel is the amount of heat developed by its combustion, expressed in B.t.u. per pound of fuel. The heat value of coal is determined by igniting a sample of known weight in a closed vessel surrounded by water and noting the rise in temperature of the water. From the previously determined thermal capacity of the vessel and water the heat developed can be computed. The calorific values of the various kinds of coal were given in Table 31.

The fusion or **ash-softening temperature** is also an important characteristic of coal. It can be determined by grinding a sample of ash and forming a small cone of it, after which it is placed in a laboratory furnace. The temperature at which the cone fuses to a spherical lump is called the ash-softening temperature. A refractory or heat-resistant ash has a softening temperature above 2600°F., one of medium fusibility softens between 2200 and 2600°, and an easily fusible ash softens below 2200°. No exact method of predicting the clinkering properties of a coal has yet been developed, but in general, the higher the ash-softening temperature, the less the possibility of the ash forming bad clinkers. The ash-softening temperature generally decreases as the percentage of sulphur increases; but the sulphur percentage is not always in exact relation to the fusion temperature, as many low-sulphur coals will also clinker.

99. Coke.—Coke is the residue left after the volatile matter is driven off from bituminous coal and consists mainly of carbon. It is rather slow to ignite but burns with a clean flame, producing neither soot or smoke. It is produced as a by-product in the manufacture of artificial gas and is also manufactured for various industrial purposes. Its bulk is so great that the fire pot will hold only a relatively small weight of fuel, which is consumed rapidly; thus frequent firing is required unless a very deep bed of fire is maintained.

Table 34 gives the analyses of coke made from three types of coal.

TABLE 34.—CHARACTERISTICS OF COKE
(U. S. Bureau of Mines)

Source of coal	Volatile matter, per cent	Fixed carbon, per cent	Ash, per cent	Sulphur, per cent	Heating value, B.t.u. per pound
Harlan Co., Kentucky..	0.70	94.55	4.75	0.48	13,558
Jefferson Co., Pennsylvania...............	1.05	85.06	13.89	0.85	12,206
Columbiana Co., Ohio..	0.98	88.34	10.68	1.00	12,746

100. Combustion.—Combustion may be defined as the chemical combination of a substance with oxygen proceeding at such a rate that a high temperature is produced. Carbon is the principal combustible in coal. When its combustion is complete, it forms carbon dioxide (CO_2); when it is incomplete, it forms carbon monoxide (CO). The hydrogen in the coal unites with oxygen to form water vapor, and the nitrogen, which is an inert substance, is set free. For economy in fuel consumption it is necessary that combustion be complete, and to this end the supply of air must be ample; yet it is practically impossible to supply just the correct amount of air needed. In order to ensure a sufficient supply to all parts of the fuel bed with hand-firing, it is necessary to supply from 150 to 300 per cent of the theoretical requirements, whereas with mechanical stokers 30 to 50 per cent excess air is usually considered sufficient. As all this excess air leaves the boiler at the flue-gas temperature, it is evident that in the interest of economy this necessary amount of excess air should be reduced to the minimum. The best index of the amount of excess air is the percentage of CO_2 in the flue gas. If exactly enough air could be supplied, the CO_2 content, by volume, of the flue gas would be approximately 21 per cent in the combustion of pure carbon. With fuels containing hydrocarbons this figure is considerably lower. Therefore the type of fuel must be taken into account when the CO_2 content of the flue gas is used as a measure of the amount of excess air. Figure 55 shows this relationship for various fuels. In hand-fired furnaces the CO_2 ranges from 5 to 13 per cent under various fuel-bed conditions but rarely goes above 10 per cent. With mechanical stokers of the type used in heating boilers it may range between 8 and 12 per cent. With oil, because of the closer control of excess air, the CO_2 should vary between 10 and 12 per cent, which means an excess air quantity of about 30 per cent.

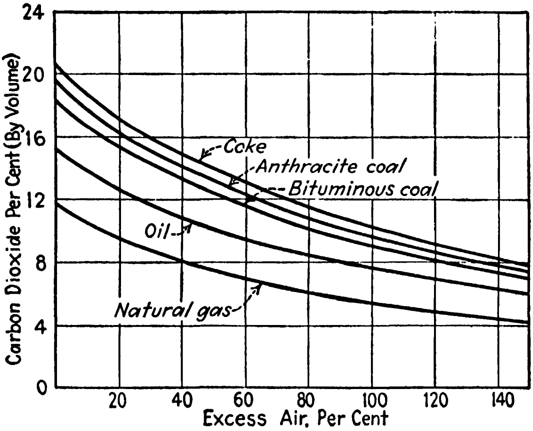

FIG. 55.—Relation of carbon dioxide in the flue gas to excess air.

The rate of combustion also depends on the temperature of the mixture and is greatly accelerated with high furnace temperatures. The ignition temperature of a coal or gas is the lowest temperature at which it will ignite. Evolution of the combustibles begins at about 650°F. The ignition temperature of the coke formed by the fixed carbon of bituminous coal is around 765°, while the minimum temperature at which the combustible gases will ignite is over 1100°.

Gases distilled from the volatile matter in coal are ignited and burn above the fuel bed, but the fixed carbon must be pushed to the hotter zone before it will ignite. If the temperature of the combustion space is below the ignition point of the gases, or if they are chilled by contacting cold heating surfaces of the boiler or furnace before becoming completely burned, incomplete combustion results, usually accompanied by smoking and the formation of soot.

Incomplete combustion also results when the air supply is deficient or is incompletely mixed with the volatile matter given off by the fuel. The presence of carbon monoxide in the flue gases is an indication of incomplete combustion.

101. Flue-gas Losses.—When there are no unburned combustibles (CO) in the flue gases, the *combustion efficiency* is generally considered to be an effective measure of fuel utilization. Combustion efficiency in percentage of the heat released by the fuel is equal to 100 minus the percentage of the total flue-gas losses. A measurement of the flue-gas temperature and CO_2 content at the smoke outlet of a boiler or furnace may be used to determine the total stack losses for coal, oil, or gas from Fig. 56.[1] These curves are useful for determining the efficiency of heating plants in the field, estimating fuel comsumption, and making combustion adjustments for economical operation. By assuming reasonable values for radiation loss from equipment and unaccounted-for losses such as unburned combustibles, an approximation of over-all heating-equipment efficiency can be estimated.

From the curves in Fig. 56 it will be noted that if a heating device does not have sufficient effective heating surface, with resulting flue-gas temperatures exceeding approximately 500°F., then any increase in CO_2 will be accompanied by large reductions in the flue-gas losses. On the other hand, if a unit is provided with sufficient heating surface for a given fuel input, so that the leaving flue-gas temperature is low, then the relative changes in the flue-gas losses will be small, even with large changes in the CO_2 content of the gases. A unit operating under the latter conditions may be expected to perform satisfactorily under diverse field applications, even though the adjustments for fuel and excess air may vary widely.

With a constant fuel rate, any decrease in excess air is accompanied by an increase in CO_2 and a lower stack temperature, resulting in a higher combustion efficiency. It is true that the flame temperature in the combustion chamber is higher with less excess air. However, the exit-gas temperature is lower, because of the improved rate of heat transfer. This improvement is caused by the fact that with a higher initial temperature more

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[1] *Univ. Ill. Eng. Exp. Sta. Circ.* 44.

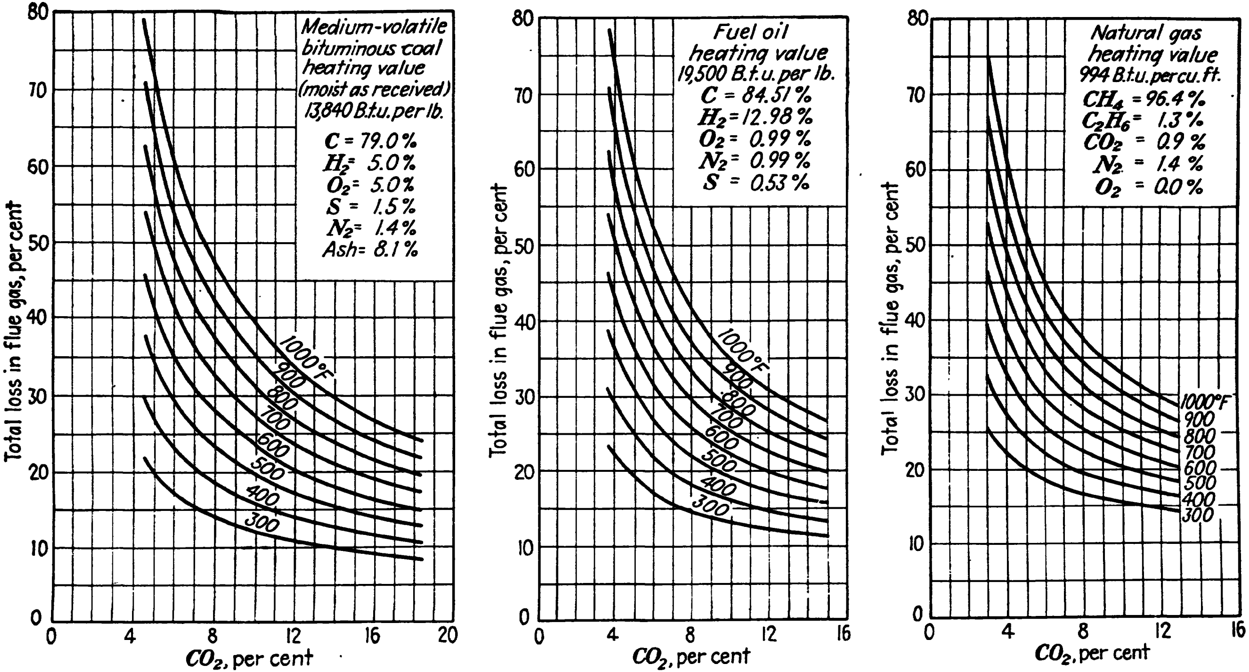

Fig. 56.—Flue-gas losses with various fuels.

heat is absorbed by radiation on the surfaces of the unit. Heating equipment designed to operate with a stack temperature above approximately 500° will improve by approximately 1 per cent in combustion efficiency for every decrease of 30° in exit-gas temperature.

Example 1.—Determine the bonnet efficiency of an oil-burning furnace if the flue-gas temperature at the smoke outlet of the unit is 500°F. and the CO_2 content is 12 per cent. Assume a radiation loss from the casing amounting to 6 per cent and unaccounted-for losses of 3 per cent.

From the oil curves in Fig. 56 it will be found that the total stack or flue-gas loss is 17.5 per cent for the stated stack temperature and CO_2. The bonnet efficiency of the furnace is then $100-(17.5 + 6 + 3)$ or 73.5 per cent.

Example 2.—Assume that a residential boiler is designed to burn either coal, oil, or gas so that the flue-gas temperature measured at the smoke outlet is 600°F. for all fuels. If the radiation and unaccounted-for losses total 10 per cent, determine the percentage of CO_2 each fuel-burning unit must produce to maintain a boiler efficiency of 65 per cent.

Added together, the boiler efficiency of 65 per cent and the miscellaneous losses of 10 per cent indicate that the fuel-burning equipment must be adjusted so that the total flue-gas loss will not exceed $100 - 75$ or 25 per cent. From the coal curve in Fig. 56 it will be found that for a stack loss of 25 per cent and a flue-gas temperature of 600°, the CO_2 should be 9.2 per cent. Similarly, for oil the CO_2 should be 8.3 per cent, and for gas 8.1 per cent.

102. Smoke.—Smoke consists principally of unburned carbon in finely divided particles set free by the splitting up of unburned hydrocarbon gases. Although the waste represented by the visible products themselves is not great, smoke is an indication of incomplete combustion and consequently of wasted fuel.

Smoke may be avoided by the use of anthracite coal, coke, or the semibituminous coals, which have little volatile matter, or by ensuring complete combustion when coals high in volatile content are used. When coal containing much volatile matter is placed on a hot bed of fuel, the volatile matter is distilled off; and in order that complete combustion of this gas may take place, sufficient air must be supplied and intimately mixed with the combustible gases. Furthermore, the combustion space must be of sufficient size that combustion can be completed before the gases come into contact with the relatively cold surfaces of the boiler. The air supply must not be so copious or at such a low temperature as to chill the mixture below the temperature required for combustion. These requirements are met by the use of various appliances and of furnaces of special design, which will be discussed later.

103. Ash and Clinker.—Ash is foreign matter in the coal, part of which is present in the coal itself, the remainder coming from above and below the seam as it is mined. Ash is objectionable because it reduces the heating value of the coal and because of the trouble that it causes in the furnace. An excessive amount of ash obstructs the passage of air through the fuel bed, causes clinker formation, and carries much unburned fuel with it into the refuse pile. Clinker is simply ash that has fused and run together. Clinker formation is most frequent and troublesome when the ash has a low melting point. A low melting point is thought to be due to the presence of excessive sulphur and of iron oxides in the ash. However, the clinkering of the ash is necessary in most stokers used for residence heating and in many commercial-sized stokers in order that the ash may be readily removed from the furnace.

A coal having an ash-fusion temperature of from 2100 to 2500°F. is usually preferred for residential stokers. Ash of a lower fusion temperature can be burned satisfactorily if there is ample grate area and the fire is cleaned at frequent intervals to prevent clinker from running into the retort and tuyères. Ash of a higher fusion temperature can be burned, but it is often necessary to restrict the grate area with refractory in order to fuse the ash into clinker for easy removal.

On large commercial stokers best results are obtained with ash-softening temperatures above 2500°, although with water-cooled tuyères and air-cooled grates satisfactory performance has been obtained where the ash-fusion temperature was as low as 1900°. The ash-softening temperature is not so critical with chain-grate, overfeed, or spreader-type stokers.

104. Comparison of Different Fuels.—The following is a summary of the advantages and disadvantages of the more common solid fuels. This comparison is made only from the standpoint of their use in heating boilers and furnaces.

BITUMINOUS COAL

Advantage:
- Low cost.

Disadvantages:
- Dirty to handle.
- Difficult to burn without smoke and soot.
- Forms clinker.

SEMIBITUMINOUS COAL

Advantages:

Low cost.
Burns with little smoke.

Disadvantage:

Dirty to handle.

ANTHRACITE COAL

Advantages:

Clean to handle.
Burns without smoke.
Maintains a steady fire with infrequent attention.

Disadvantages:

High cost.
Sometimes high in ash content.

COKE

Advantages:

Fairly clean to handle.
Burns without smoke.
Moderate cost.

Disadvantages:

Requires frequent firing.
Difficult to maintain a steady fire.

105. Oil.—Oil has in the last few years taken its place as a fuel of major importance for heating systems, and automatic and semiautomatic oil burners have been brought to a satisfactory state of mechanical perfection.

Crude petroleum is a mixture of hydrocarbons of varying volatility and contains also small amounts of sulphur, nitrogen, and oxygen. An average oil contains approximately 85 per cent of carbon and 12 per cent of hydrogen by weight. The behavior of oil as a fuel, however, depends upon its physical characteristics rather than its chemical analysis.

Crude petroleum is seldom used as a fuel but is broken up by distillation into various grades of oil, which are used for different purposes. The first stages of the distillation process produce gasoline and naphtha. Then come the various grades of fuel oil of increasing specific gravity, and finally a heavy residue is left.

As a result of the introduction of high-octane gasoline, it is expected that future domestic fuel oils will be produced by the new catalytic processes. These oils will be somewhat lower in gravity and will have a higher heat content, which means that more intimate mixing of the air and oil will be required if the burner is to operate without excessive smoke.

Oils are classified by their specific gravity. The scale commonly used is that of the American Petroleum Institute which is an arbitrary one derived from the Baumé scale and determined as follows:

$$\text{A.P.I. gravity} = \frac{141.5}{\text{specific gravity}^{1}} - 131.5 \qquad (1)$$

It is evident that the heavier the oil, the lower its A.P.I. gravity.

The heating value of oil fuel, that is, the amount of heat developed by complete and perfect combustion, varies between 18,500 and 19,500 B.t.u. per lb. and between 135,000 and 155,000 B.t.u. per gal. The chart in Fig. 57 shows the relation between heating values and specific gravity.

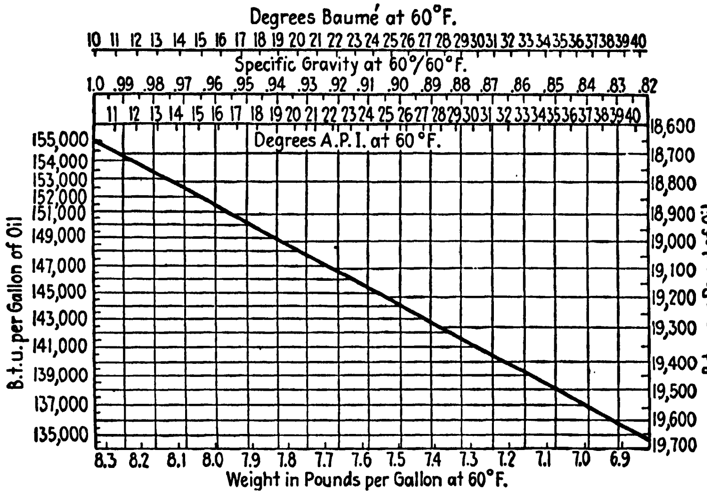

FIG. 57.—Relation between specific gravity and heat content of fuel oils. (*American Oil Burner Association*, "*Handbook of Domestic Oil Heating*.")

The lighter oils, that is, those having the higher A.P.I. gravity, are the easier to handle in a burner but are also the more expensive. Oils of gravity below 24°A.P.I. usually require preheating to reduce their viscosity.

106. Oil Specifications.—Gravity alone is not a sufficient or reliable index of the properties that are important, although some of them are related to the gravity.

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[1] Determined at 60°F.

TABLE 35.—FUEL-OIL STANDARDS[1]
(National Bureau of Standards CS 12-40)

Grade[2]	Flash point, °F.		Pour point, °F.	Water and sediment, per cent	Carbon residue, per cent	Ash, per cent	Distillation temperatures, °F.				Viscosity seconds				Heating value and gravity[3]	
							10 Per cent point	90 Per cent point		End point	Saybolt Universal at 100°F.		Saybolt Furol at 122°F.		Approx. heating value, B.t.u. per gal.	A.P.I. gravity
	Min.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Min.	Max.	Max.	Min.	Max.	Min.		
No. 1 Fuel oil—a distillate oil for use in burners requiring a volatile fuel.	100 or legal	165	0[4]	Trace	0.05 on 10% residuum[5]		410			560[6]					136,000	36°-40°
No. 2 Fuel oil—a distillate oil for use in burners requiring a moderately volatile fuel.	110 or legal	190	10[4]	0.05	0.25 on 10% residuum[7]		440	600							138,500	32°-36°
No. 3 Fuel oil—a distillate oil for use in burners requiring a low viscosity fuel.	110 or legal	230	20[4]	0.10	0.15 straight			675	600[8]		45				141,000	28°-32°
No. 5 Fuel oil—an oil for use in burners requiring a medium viscosity fuel.	130 or legal			1.00		0.10						50	40		149,000	18°-24°
No. 6 Fuel oil—an oil for use in burners equipped with preheaters permitting a high viscosity fuel.	150			2.00[9]									300	45	150,000	12°-18°

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[1] Recognizing the necessity for low sulphur fuel oils used in connection with heat-treatment, nonferrous metal, glass and ceramic furnaces, and other special uses, a sulphur requirement may be specified in accordance with the following table:

Grade of Fuel Oil	Sulphur, Max. Per Cent
No. 1	0.5
No. 2	0.5
No. 3	0.75
No. 5	No limit
No. 6	No limit

Other sulphur limits may be specified only by mutual agreement between the buyer and seller.

[2] It is the intent of these classifications that failure to meet any requirement of a given grade does not automatically place an oil in the next lower grade unless in fact it meets all requirements of the lower grade.

[3] Not a part of Standard CS 12-40.

[4] Lower or higher pour points may be specified whenever required by conditions of storage or use. However, these specifications shall not require a pour point lower than 0°F. under any conditions.

[5] For use in other than sleeve-type blue-flame burners, carbon residue on 10 per cent residuum may be increased to a maximum of 0.12 per cent. This limit may be specified by mutual agreement between the buyer and seller.

[6] The maximum end point may be increased to 590°F. when used in burners other than sleeve-type blue-flame burners.

[7] To meet certain burner requirements the carbon residue limit may be reduced to 0.15 per cent on 10 per cent residuum.

[8] The minimum distillation temperature of 600°F. for 90 per cent may be waived if A.P.I. gravity is 26 or lower.

[9] Water by distillation, plus sediment by extraction. Sum, maximum 2.0 per cent. The maximum sediment by extraction shall not exceed 0.50 per cent. A deduction in quantity shall be made for all water and sediment in excess of 1.0 per cent.

The **flash point** and the fire point are particularly important as indications of the performance of an oil in a burner. The flash point is the temperature at which the oil becomes sufficiently volatile so that the vapor will ignite with a flash. The test is made by slowly heating a sample of the oil in a closed cup[1] surrounded by an air bath, which is heated by a Bunsen burner. From time to time a flame is introduced into the cup, and the temperature of the oil at which the first flash of ignited vapor occurs is the flash point.

An oil with a low flash point ignites readily in an oil burner; this factor is especially important in a burner of the intermittent type, but too low a flash point increases the hazard in storing and handling.

The **fire point** is determined by means of the same apparatus and is the lowest temperature at which the oil burns with a steady flame instead of merely flashing and becoming extinguished. The fire point is usually about 20° above the flash point.

For oil burners provided with automatic ignition, a flash point of 100 to 150°F. is desirable. An oil having a flash point of over 170° is difficult to ignite by the automatic means ordinarily provided but may be used in a continuously operated burner.

The next most important points are the **distillation characteristics,** which influence the ignition and burning qualities of an oil. It may have a sufficiently low flash point through the presence of a small quantity of high-volatile oil but may be otherwise unsatisfactory. The distillation characteristics are expressed as the temperatures at which certain percentages (usually 10 and 90 per cent) of the oil are distilled off when the oil is subjected to complete distillation.

A balanced fuel oil is one that combines the correct proportions of light, intermediate, and heavy fractions to provide the necessary elements for instantaneous ignition, smooth, progressive volatilization, and clean, efficient combustion.

The **pour point** is the lowest temperature at which the oil will flow. It is important because oil is usually stored in outdoor tanks either above or below ground.

The **viscosity** of an oil is an index of its resistance to flow and is a very important characteristic. A low viscosity is necessary for an oil to flow freely and to atomize properly in the burner. Oils of high viscosity require preheating to make them usable.

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[1] Flash Point by Means of the Pensky-Martens Closed Tester, A.S.T.M. Designation D 93-42.

Pressure-atomizing burners of low capacity require an oil of uniform viscosity, because a change in viscosity will alter the burning rate and thereby alter the correct mixture of oil and air for perfect combustion. The viscosity of an oil not only affects the output of a nozzle but also changes the angle of spray.

Viscosity is usually determined by measuring the time required for a measured quantity to flow through an orifice and is expressed on the so-called "Saybolt scale." Since the viscosity decreases as the temperature rises, it is necessary to conduct the test at a definite temperature.

Both pot and rotary vaporizing-type burners are sensitive to oils having high *carbon residue*, which is the solids remaining after the fuel is burned. Oils having a high carbon residue cause accumulations of carbon to form on the vaporizing surfaces when subjected to the necessary high temperatures.

The National Bureau of Standards has a set of standard specifications for domestic and industrial fuel oils, which are shown in Table 35. The oil-burner manufacturer knows the oil best suited to his particular burner, inasmuch as a burner is or should be designed for a definite grade of oil.

Tentative standards covering the new catalytic fuel oils are now being developed by Committee D-2 on Petroleum Products of the American Society for Testing Materials.

107. Combustion of Oil.—The combustion of oil can take place only after it has been gasified. Gasification is accomplished principally by heating but may be accelerated by atomization, which is simply mechanical breaking up of the oil into very small globules. Since gasification takes place from the surface of the oil, atomization accelerates it by enormously increasing the amount of surface, permitting the radiant heat from the flame or from hot brickwork to strike it.

The gasified oil burns when raised to a sufficient temperature in the presence of air, and the carbon and hydrogen unite with the oxygen of the air in a series of successive reactions to form (if the combustion is complete) carbon dioxide and water. If the combustion is not completed, owing to an insufficient air supply or to the chilling of the flame by contact with cold surfaces, carbon will be deposited in the form of soot. The combustion space must be amply large and unobstructed to allow complete combustion.

The amount of excess air being supplied to an oil flame can best be determined by means of a flue-gas analysis but may be roughly gaged by observing the character of the flame. If the amount of excess air is high, the flame will be almost white in color; and as the amount of air is reduced, the flame will change to yellow and then to orange and finally to a red, the last usually being a smoky flame. In adjusting the air supply, a bright orange color is usually the most satisfactory, indicating an excess of air of approximately 25 per cent. A blue flame is obtained in vaporizing-type burners when the oil is vaporized and mixed with air before ignition. It does not necessarily indicate perfect combustion.

Industry standards[1] have been adopted which provide that a domestic oil burner should produce a combustion efficiency as measured by the percentage of CO_2 in the flue gases of not less than 10 per cent in the laboratory and 8 per cent in the field without producing smoke.

108. Oil Burners.—When oil is burned under a boiler, it is necessary to provide some device to atomize or vaporize the oil and to feed the burning jet at a uniform rate. Most oil burners are designed to atomize the oil, and vaporization takes place after atomization. There are satisfactory burners, however, which vaporize the oil by distilling it upon a hot surface or in a heated precombustion chamber. Since the oil is gasified, flame length is short and combustion-chamber volumes can be kept small. Vaporizing burners require a light grade of oil for good results and must be carefully installed and operated to avoid trouble from carbon deposits.

Atomization is accomplished by means of steam, air, or centrifugal force. Figure 58 illustrates one type of steam-atomizing nozzle in its elementary form. The steam jet, in this case of annular shape, strikes the oil stream at right angles, tearing the oil to pieces and projecting it into the combustion space. A nozzle such as this, properly installed in a suitable combustion chamber, constitutes an oil-burning system; but usually in heating work it is also necessary to provide more or less complete automatic control, and only in high-pressure plants is steam or air available at high pressure. Therefore oil burners for heating work are usually built more like the form shown in Fig. 59 and include a blower for supplying air for atomization and part or all of the air for combustion, and an oil pump. Atomization is effected in a mixing nozzle. This is commonly called the "gun type" or "high-pressure atomizing type" of burner.

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[1] Automatic Mechanical Draft Oil Burners Designed for Domestic Installations, National Bureau of Standards, Commercial Standard CS 75-42.

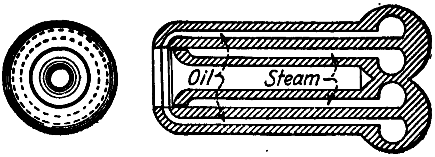

Fig. 58.—A simple steam-atomizing nozzle. Notice that the steam is in the inner chamber and the oil in the outer chamber. This "rose" burner makes a round flame. (*"Handbook of the Petroleum Industry,"* *Day.*)

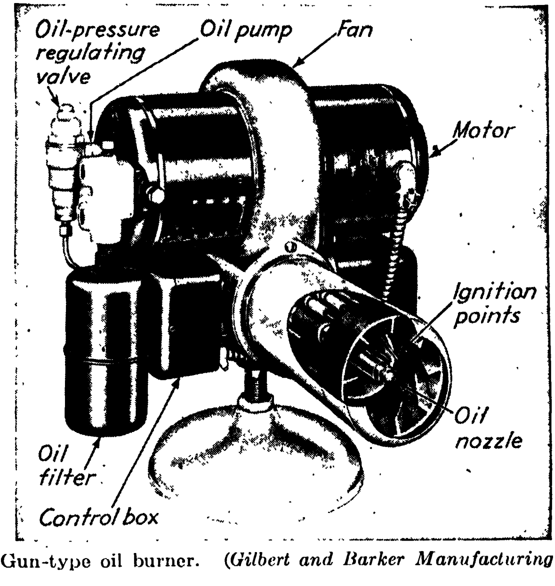

Fig. 59.—Gun-type oil burner. (*Gilbert and Barker Manufacturing Company.*)

Mechanical atomization by centrifugal force is the principle of operation of the burner illustrated in Fig. 60. The oil is forced up through the center of the hollow vertical shaft and projected radially from the edge of the revolving disk by centrifugal force.

Most of the domestic oil burners on the market, excepting the vaporizing type, are of the gun type of Fig. 59 or the centrifugal type of Fig. 60.

For larger buildings oil burners of larger capacity, more rugged and capable of burning the heavier grades of oil, are used. Figure 61 shows a burner of this class, designed to be mounted on the boiler front. An oil heater, usually a tube-type heater connected to the water space of the boiler, is required for the heavy oils.

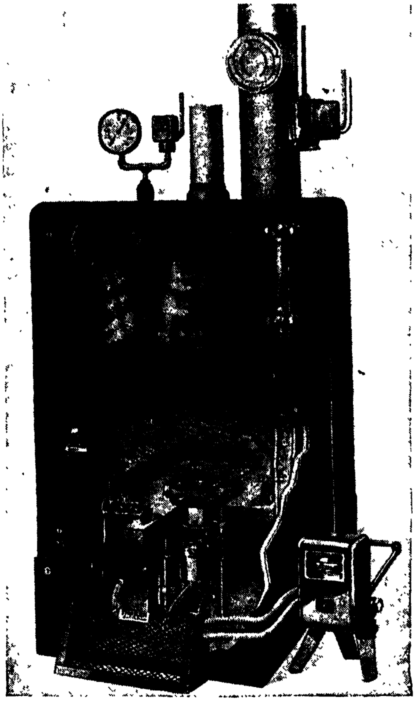

FIG. 60.—Centrifugal-type oil burner. (*Timken Silent Automatic Division, Timken-Detroit Axle Company.*)

It is desirable to choose an oil burner that gives a flame approximately the shape of the combustion chamber of the boiler in which it is to be installed. It is undesirable to have the flame impinge upon the cold boiler surfaces or upon unprotected metal. Refractory lining should be provided where necessary.

The provisions for the handling and storage of oil are important, particularly from the standpoint of safety. The National

Board of Fire Underwriters permits the use of a basement tank of limited capacity. Tanks of larger size must be installed outside and are usually placed below ground. The oil is delivered to the burner by a pump either wall-mounted or attached to the burner.

109. Oil-burner Ignition and Control.—Automatic ignition is provided when a burner is designed for intermittent operation. Gas, oil, and electricity are used in various ways for ignition. In large installations with continuous attendance, the burners may be arranged to operate continuously with a flame of varying size, but in small installations full on-or-off operation is generally used.

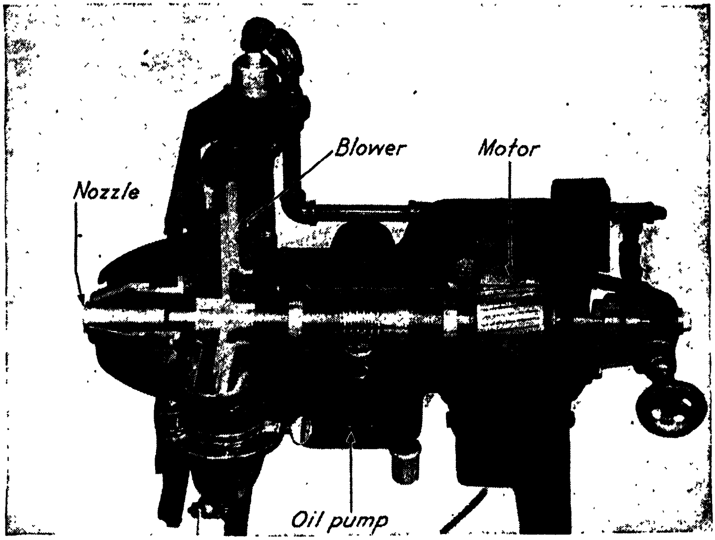

Fig. 61.—Oil burner for large installations. (*Enterprise Oil Burner Company.*)

The automatic control of an oil burner is an important part of the installation. In residence work a central thermostat starts and stops the burner, and good regulation may be obtained if the thermostat is properly located. Safety devices of various kinds are a part of the control system. In a steam heating system controls are needed (*a*) to protect the boiler against overpressure or (*b*) low water and (*c*) to shut off the burner in case the oil for any reason fails to ignite. It is becoming generally recognized that all three of these protective features are quite necessary.

The protection against overpressure is usually accomplished by a contact-making pressure gage, attached to the boiler, which functions to stop the burner in case of excess pressure. The low-water cutoff is simply a float-operated switch, which opens when the water level in the boiler drops below a predetermined point and stops the burner. The latter is often combined with an automatic water feeder, which admits water to the boiler before the water level drops to the danger point at which the cutoff operates. A device of this kind is shown in Fig. 62.

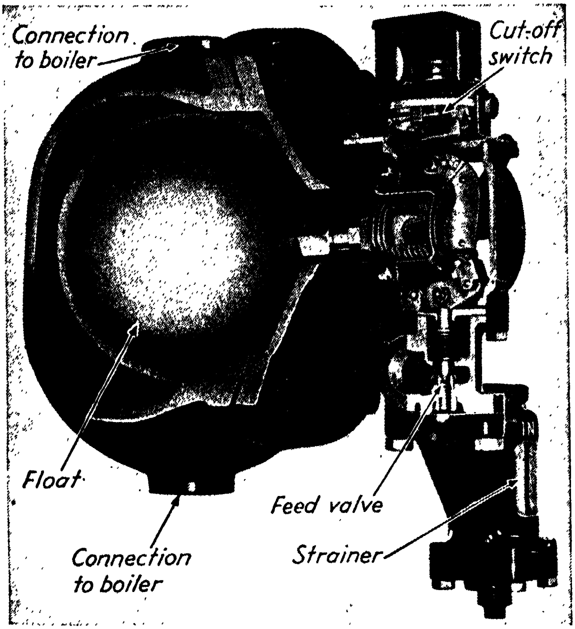

Fig. 62.—Safety feeder and low-water cutoff. (*McDonnell and Miller Company.*)

The protection against failure of the burner to ignite is accomplished in several ways. One common arrangement makes use of a thermal element in the smoke pipe. In normal operation the temperature of the stack gases rises very soon after the burner starts. If for any reason the oil does not ignite, and the thermal element does not indicate within a few seconds that heat has reached it, the burner is automatically cut off.

In a gravity warm-air furnace system the safety devices usually comprise (*a*) protection against excessive bonnet temperature, accomplished by a thermostat placed in the bonnet, and (*b*) protection against failure of the burner to ignite, which is accomplished by installing a thermal element in the smoke pipe. In a mechanical warm-air system these controls are also used, plus a means of controlling the fan, which will be described later.

There are many different ways in which these various safety devices may be arranged, and there are some very ingenious and more or less complicated control systems in use, but all of them are designed to accomplish the purposes stated above.

TABLE 36.—COMPOSITION OF ARTIFICIAL GAS, PER CENT BY VOLUME

Constituent	Carbureted water gas	Coke-oven gas	Water gas
Hydrogen	40.0	50.0	48.0
Methane	25.0	36.0	2.0
Ethylene	8.5	4.0	
Nitrogen	4.0	2.0	5.5
Carbon monoxide	19.0	6.0	38.0
Oxygen	0.5	0.5	0.5
Carbon dioxide	3.0	1.5	6.0
B.t.u. per cu. ft.	575.0	603.0	295.0

110. Gas.—The use of artificial gas as a fuel for house heating is rapidly increasing. The simplicity of the apparatus required to burn gas, as compared with the oil burner, and the somewhat greater reliability are attractive to the consumer. The cost of artificial gas is in most instances greater than that of oil or solid fuel.

Artificial gas contains from 500 to 575 B.t.u. per cu. ft., value of 530 B.t.u. being specified by law in many cities. It consists of a mixture of carbon monoxide, hydrogen, and methane in proportions that depend upon the method of manufacture. Typical analyses of the three principal kinds of artificial gas are given in Table 36. The desired B.t.u. value is often obtained by blending different kinds of gas.

Natural gas has recently become an important fuel. As found, it consists almost entirely of methane and ethane, with a heating value of 900 to 1200 B.t.u. It is often re-formed so as to have a heating value of 530 to 550 B.t.u. and then contains some hydrogen. Its cost is usually low and in some localities compares very favorably with that of other fuels.

Most gas burners are of the Bunsen type and operate with a nonluminous or blue flame. Combustion is effected by mixing primary air with the gas ahead of the burner, and secondary air is introduced around the flame by draft inspiration. In the luminous- or yellow-flame burner, all the air for combustion is brought in contact with the flame as secondary air. Furnace volumes may be small compared with coal or oil, owing to the short flame length.

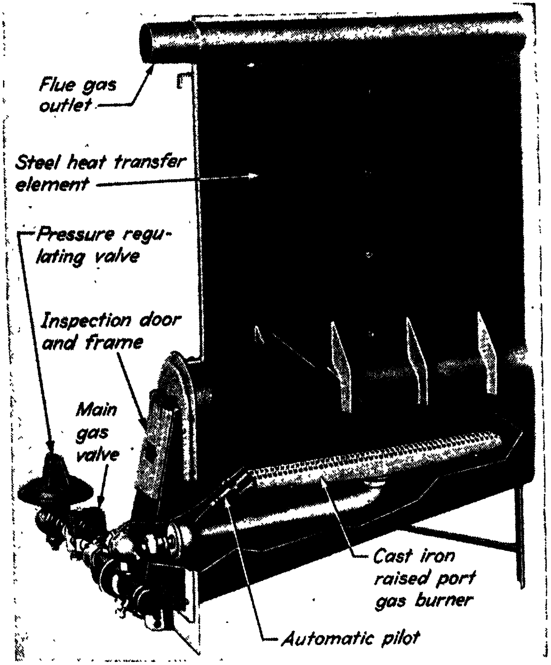
Flue gas outlet
Steel heat transfer element
Pressure regulating valve
Inspection door and frame
Main gas valve
Cast iron raised port gas burner
Automatic pilot

FIG. 63.—Gas-fired, warm-air furnace section. (*Waterman-Waterbury Company.*)

One heat-transfer section of a furnace designed for gas is illustrated in Fig. 63, complete with a cast-iron Bunsen type of burner with raised port, together with the necessary safety devices and control valves. Furnaces of this type are arranged with multiples of these sections in either a gravity or forced warm-air design to produce the necessary heat output.

Standards[1] stipulate that these furnaces operate under laboratory test with a minimum over-all efficiency of 70 per cent for the gravity type and 75 per cent with a fan or forced-air arrangement. The required efficiency for steam or hot-water boilers is 75 per cent. The flue-gas temperature for domestic boilers is limited to 480°F. above room temperature.

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[1] Approval Requirements for Central Heating Gas Appliances, American Standards Association, Z21.13-1943.

111. Stokers.—Stokers used in heating work for burning bituminous coal are principally of the underfeed type such as illustrated in Fig. 64. The coal is fed by a screw or a ram from the hopper into a retort that is bordered by tuyères. A centrifugal fan delivers the combustion air to the tuyères, and the combustion in and above the retort takes place at a high rate. The ash accumulates in the form of clinkers on the dead plates surrounding the retort and is removed through the furnace door.

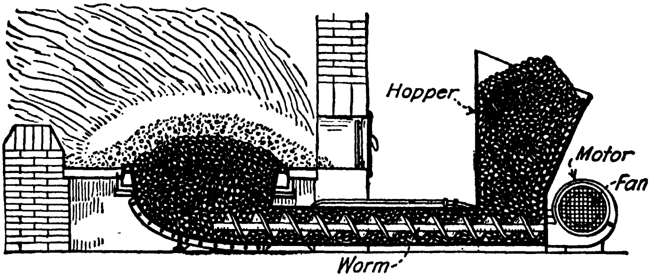

Fig. 64.—Automatic underfeed stoker.

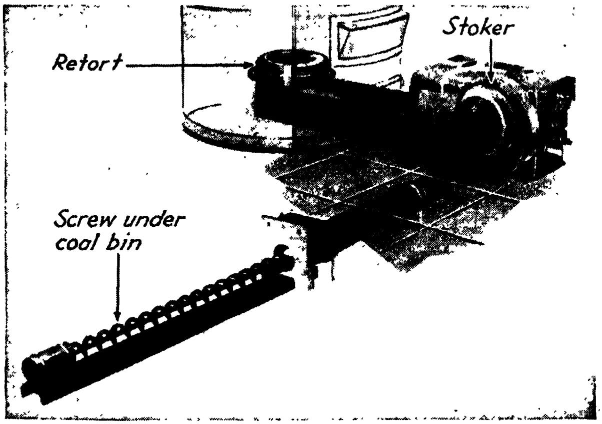

Fig. 65.—Stoker with bin feed. (*Schwitzer-Cummins Company.*)

To eliminate the labor of filling the hopper, a second screw may be provided to transport the coal from the bin to the stoker. A stoker thus equipped is shown in Fig. 65. Some residential stokers are designed to remove the ashes automatically and deposit them in an ash receptacle.

At the present time about 70 per cent of the number of stokers installed yearly are of the residence type with capacities up to 60 lb. of coal per hr. This field is developing rapidly because of the demand for economical automatic heat, and the stokers of that type are being constantly improved.

Recently several lines of boilers have been designed especially for stoker firing, but the majority of installations require care in adapting the stoker to the boiler. The furnace volume and the furnace height and shape are the important factors.

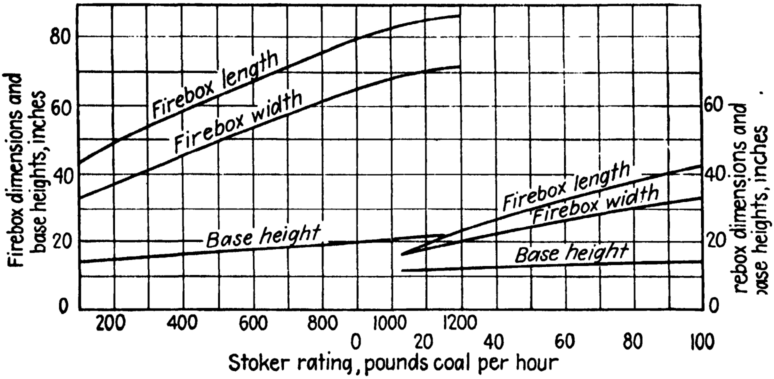

FIG. 66.—Suggested minimum firebox dimensions and base heights.[1]

Standards[2] for minimum firebox dimensions and base heights for stokers have been formulated according to the curves shown in Fig. 66. Also, empirical formulas have been developed for determining stoker-setting heights, as follows:

For burning rates up to 100 lb. coal per hr.

$$H = 0.1125C + 15.75 \tag{2}$$

For burning rates from 100 to 1,200 lb. coal per hr.

$$H = 0.03C + 24 \tag{3}$$

where H = minimum setting height, in., for steel boilers. For cast-iron boilers height may be $\frac{7}{8}H$.

C = coal-burning rate, lb. per hr.

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[1] For reference in selecting or designing boilers and furnaces for stoker firing. Dimensions shown are for net inside clearance at grate level using coal with heating value of not less than 12,000 B.t.u. per pound. Under certain conditions smaller fireboxes will permit satisfactory performance but these dimensions are preferred normal minimums.

[2] Technical Manual, Stoker Manuacturers Association, 1944.

These dimensions should not be considered as arbitrary values for stoker settings, but they do represent average minimum requirements. Many factors such as type of stoker, quality of coal, and chimney size or height affect the proper application of stokers, and in certain instances the above dimensions may be modified without impairing performance.

The required furnace volume may be determined by multiplying grate area by the minimum permissible furnace height, but it should not be so small as to give an excessive heat release per cubic foot. Heat release per cubic foot is the total B.t.u. value of the quantity of coal to be burned per hour divided by the furnace volume. This figure should not usually exceed 50,000 B.t.u. per cu. ft. per hr. for bituminous coal. With anthracite coal, which has little volatile matter, a much smaller combustion chamber and consequently a higher heat release per cubic foot is permissible, and this is not a limiting factor.

112. Stoker Controls.—Usually the reason for installing a stoker is to obtain an automatic heating system, and this requires that the controls be complete and carefully installed. Stoker controls for residence-type units usually consist of the following:

1. A temperature control, operating from the heated space and starting or stopping the stoker as required.
2. A limit control for safety. This is usually a pressure control in a steam system, a boiler-water temperature control in a hot-water system, or a bonnet temperature control in a warm-air system.
3. A low-water cutoff.
4. A hold-fire control. During warm days a stoker fire may die out unless started up periodically. The hold-fire control automatically starts up the stoker and fan, say, once an hour and operates them for a short period to revive the fire.

113. Combustion in a Small Stoker.—The combustion process in the small single-retort stoker is influenced to a large extent by the type of coal burned, and perhaps the most important property of the coal is its coking characteristic. This is well illustrated in Fig. 67, which shows the cross section of two stoker fuel beds, one with a strongly coking coal and one with a weakly coking coal. A strongly coking coal has a tendency to form a coke spire, or tree, which may cause considerable difficulty. The solid mass of coke is slow to respond when the thermostat calls for heat and may even grow to such proportions as to reach the crown sheet of the boiler. Some stokers are therefore built with an agitator that prevents this formation.

The coal for small stokers must be of small enough size to feed readily. It is not usually essential that the fines be removed, because with many coals the feeding screw usually crushes the coal considerably. However, if the fines have been removed the coal will burn with less coking, and less fly ash will be liberated with the products of combustion. The size known as 1¼ × 0 is commonly used. Coal treated with oil to reduce dust is generally recommended for domestic installations.

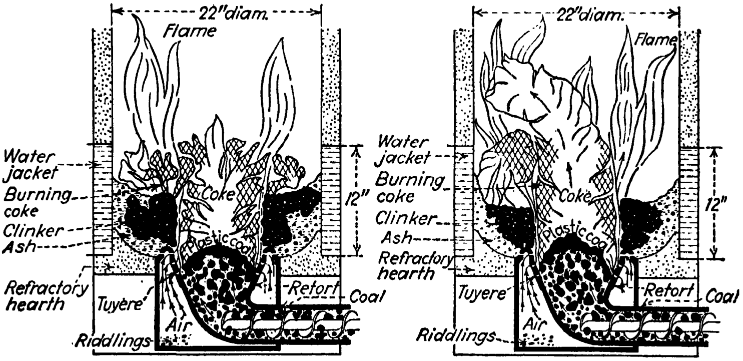

FIG. 67.—Cross sections of fuel beds in a small stoker.[1] Left, weakly coking coal. Right, strongly coking coal showing coke-tree formation.

Under intermittent control the condition of the fire varies greatly through the cycle, and the air supply should be set for the best average results rather than on the basis of snap readings of the CO_2. Most stokers are equipped with an air-control damper, which maintains an approximately constant flow of air regardless of changes in fuel-bed resistance.

For large installations requiring a coal-burning rate of 300 lb. per hr. or more, various designs of stokers are manufactured of which that shown in Fig. 68 is one example. The coal is fed into the retort by a motor-driven plunger, and auxiliary pushers propel it along the relatively narrow retort. Ashes are removed by means of side dump plates.

[1] From R. A. SHERMAN, "Automatic Residential Heating with Bituminous Coal," 22d Fuel Engineering Conference, Appalachian Coals, Inc.

One form of overfeed flat-grate stoker is the spreader type shown in Fig. 69, in which coal is distributed by air over the entire grate surface. This type of stoker may be adapted to a wide range of fuels and has wide applications on small-sized fuels, lignites, high-ash coals, and coke breeze.

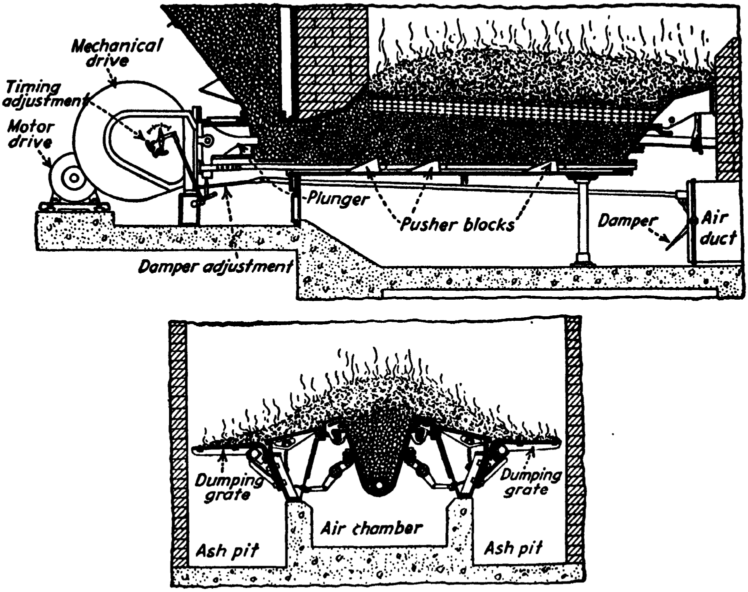

FIG. 68.—Underfeed stoker for large installations.

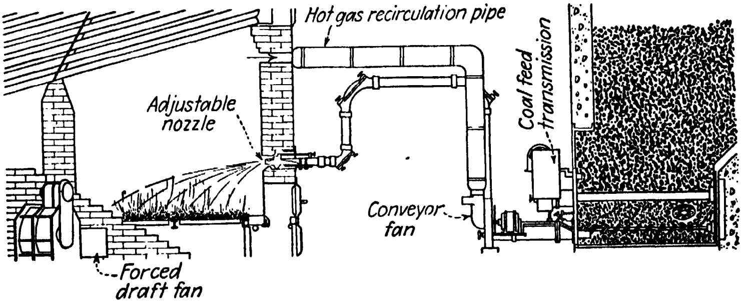

FIG. 69.—Pneumatic spreader-type stoker. (*Iron Fireman Corporation.*)

114. Selection of Stoker Size.—The Stoker Manufacturers Association has adopted a standard method of selecting stokers which gives, by means of charts and tables, the proper coal-burning rate of the stoker for the required load. It is based on an assumed over-all stoker and boiler efficiency of 65 per cent for burning rates up to 100 lb. per hr. and 70 per cent above 100 lb. An allowance of 33 per cent for the piping and pickup loads is added to the load of the actual radiator surface.

Thus for a steam heating system with 9,000 sq. ft. of installed radiator surface the capacity of the stoker burning 14,000 B.t.u. coal should be

$$\frac{9{,}000 \times 240 \times 1.33}{14{,}000 \times 0.70} = 293 \text{ lb. per hr.}$$

The factor 1.33 gives an allowance for the piping loss and for the additional capacity required for starting up.

115. Boilers.—The important considerations in the selection of a boiler for heating service are the choice of the proper *type* from a standpoint of the fuel to be burned and the nature of the service, and the selection of the proper *size*. Three principal types of boilers are used in heating work: the cast-iron boiler, the steel fire-tube boiler, and the steel watertube boiler. There are other special types.

The cast-iron boiler is widely used in residences and buildings of moderate size. The round cast-iron boiler, shown in Fig. 70, is manufactured in capacities up to 1,600 sq. ft. of radiator surface. It consists of three to five main castings such as *A*, *B*, and *C*. The castings are joined by the tapered nipples, *N*, *N* and are drawn and held together by vertical bolts. For a boiler of a given diameter, the amount of heating surface can be varied by the size or number of the intermediate sections such as *B* in the figure. Naturally the taller boilers are somewhat the more efficient, since their ratio of heating surface to grate area is greater.

The relative amounts of heat absorbed by the successive sections of a round boiler similar to that in Fig. 70 have been determined by the Anthracite Institute.[1] The average figures for tests with anthracite, coke, petroleum coke, gas, and oil are given in Table 37. There is not a great difference for the different fuels.

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[1] JOHNSON and MULCEY, "Relative Absorption of the Various Sections of a Round Heating Boiler," *Trans. A.S.H. & V.E.*, 1936.

Table 37.—Relative Amounts of Heat Absorbed by Each Section of a Round Boiler

	Heating surface, sq. ft.	Per cent of heat absorbed
Fire-pot section	13.5	56.9
Crown sheet	11.3	17.9
First intermediate section	11.0	13.1
Second intermediate section	11.0	7.9
Dome section	7.0	4.2
Total	53.8	100.0

The fire-pot section is evidently the most important, but the efficiency is appreciably improved by the intermediate sections.

The "square" sectional boiler, shown in Fig. 71, is obtainable in rated capacities up to about 18,000 sq. ft. of radiation. It consists of five to ten sections joined with nipples. In the larger sizes the sections are made in halves, to make the boiler capable of being easily transported and erected. One of the advantages of sectional boilers is that they can be brought into an existing building through a comparatively small opening.

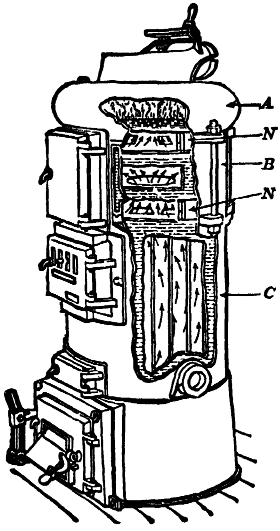


Fig. 70.—Round cast-iron boiler.

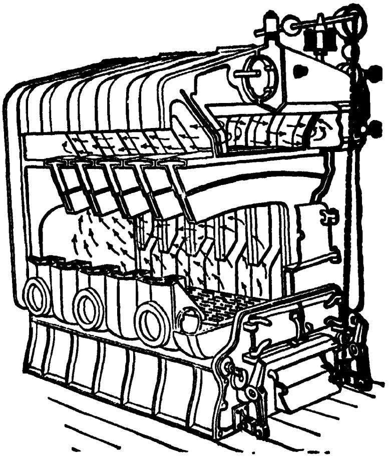


Fig. 71.—Square sectional cast-iron boiler.

The boilers shown in Figs. 70 and 71 are illustrative of the two commonest types, of which many thousands have been installed in past years. They are now being superseded to some extent by boilers of improved design and smarter external appearance, particularly where oil or gas is used as fuel.

The steel-jacketed boiler shown in Fig. 72 is an ordinary sectional boiler with an enameled steel jacket, designed to give a better appearance for residence use than a boiler covered with ordinary insulating cement. This type of boiler presents an attractive appearance and is satisfactory provided the insulation beneath the steel jacket is ample. Jacketed boilers are becoming rather widely used.

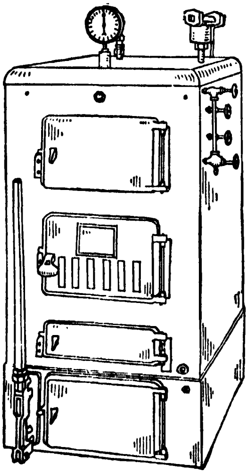


FIG. 72.—Jacketed boiler.

The design of cast-iron heating boilers is regulated in most states by the Cast-iron Boiler Code of the A.S.M.E.

116. Steel Boilers.—Boilers made of steel and assembled by means of fusion welding are quite common. They have the advantage of being less costly to build than a riveted boiler and are lighter in weight and occupy less space than a cast-iron boiler. A welded-steel boiler with the furnace incorporated with it is shown in Fig. 73. This boiler is of the fire-tube type, the flue gases passing through the tubes, which are surrounded by water. In the watertube type the water fills the tubes, which are surrounded by the hot gases. The fire-tube boiler is more common in small sizes, up to 100 hp., but large boilers are invariably of the watertube type.

Figure 74 shows a riveted-steel firebox boiler which has the furnace incorporated with it, as distinguished from the boiler in Fig. 75 in which the furnace is built up of brickwork. The latter is the well-known horizontal return-tubular boiler, so named because the gases flow toward the front of the boiler after passing around the shell. Many of these boilers are in use, and they are quite satisfactory for heating work where a relatively large capacity is required. They are suitable for working pressures of 100 lb.

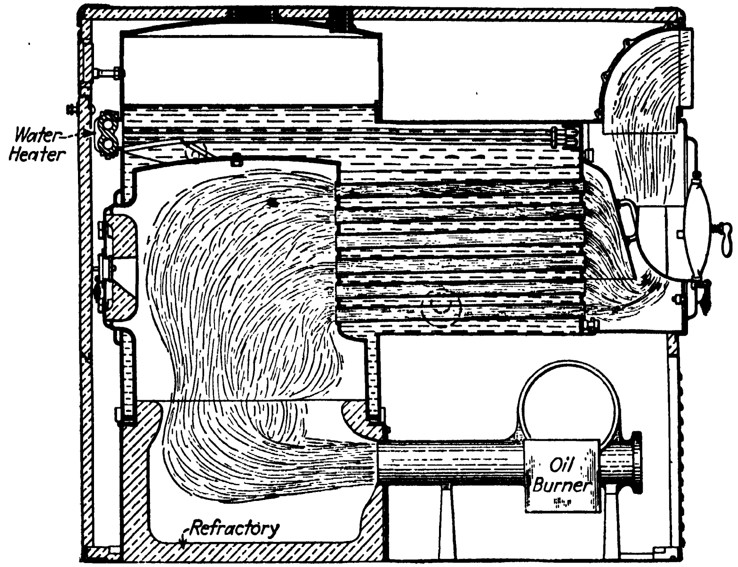


FIG. 73.—Welded-steel boiler. (*Fitzgibbons Boiler Company, Inc.*)

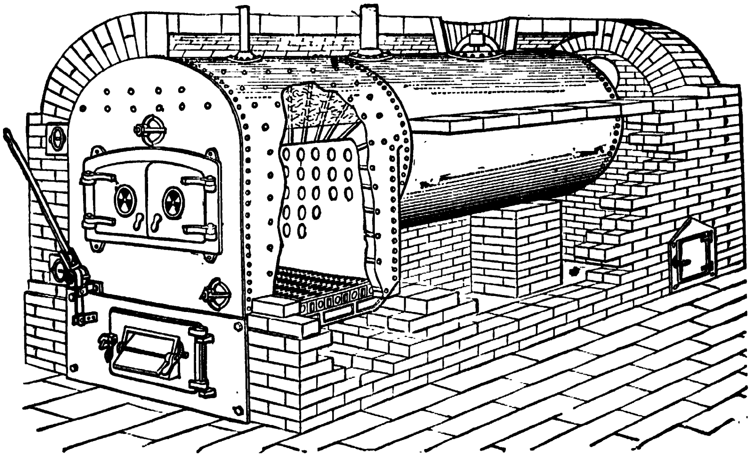


FIG. 74.—Firebox boiler.

117. Combination Units.—There is a decided trend toward the combination oil-burner-and-boiler unit built by the same manufacturer and of coordinated and compact design. Many difficulties have arisen in the past from lack of coordination

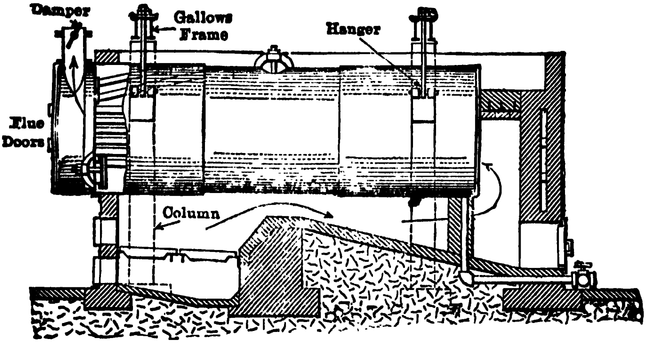


FIG 75.—Horizontal-return-tubular boiler.

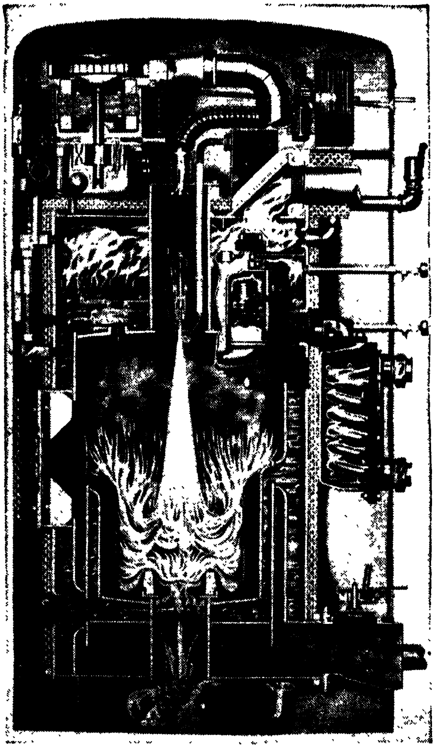


FIG. 76.—Combination oil-burner-and-boiler unit. (*General Electric Company.*)

between the two elements and from divided responsibility. The designs of combination units vary widely, but they all endeavor to meet the basic requirements of (*a*) a combustion chamber of proper design for the flame shape and (*b*) properly arranged and adequate heating surface.

Figure 76 shows one design in which the advantages of the combined unit are evident.

118. Smokeless Boilers.—It is impossible to burn bituminous coal in an ordinary hand-fired furnace without creating a certain amount of smoke. Many different furnaces have been designed to avoid smoke. In most of these, secondary air is admitted at certain points in the path of the gases and caused to mix thoroughly with them, promoting complete combustion. One of the many designs is shown in Fig. 77. Secondary air is introduced at *A* and at *B*. The fuel is introduced at the front of the furnace, and the volatile matter, as it is driven off, passes under the curtain section *B* and over incandescent fuel. On large-size boilers a brick refractory wall is frequently built at *C*, which becomes highly heated and assists in the secondary combustion process that takes place in the spaces *D*, *D*. Furnaces of this general type, although not entirely smokeless, permit the use of high-volatile fuels without violation of smoke ordinances and therefore are rather commonly used.

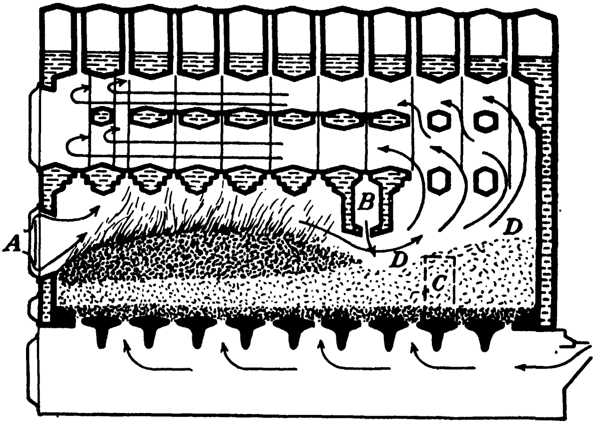
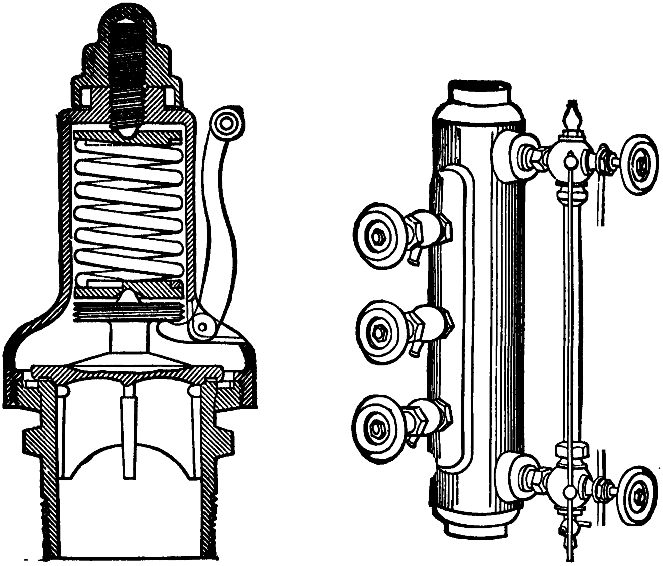


FIG. 77.—Smokeless furnace.

119. Boiler Accessories.—The two essential accessories of a steam boiler are the safety valve and the water glass. A common form of safety valve is shown in Fig. 78*a*. It is normally held closed by the spring and should be set to function at 5 to 10 lb. above the normal pressure but not over 15 lb. pressure for a cast-iron boiler.

A water glass is required to indicate the level of the water in the boiler. It may be attached directly to the boiler or, as is usual with larger boilers, fitted to a water column (Fig. 78*b*) connected by short pipes to the boiler.



(*a*) Safety valve. (*b*) Water column.

FIG. 78.—Boiler accessories.

120. Proportions of Boilers.—The heating surfaces in a boiler are defined as those surfaces that have water on one side and hot gases on the other. In order that the boiler may be efficient, the ratio of heating surface to grate surface should be large. The ratio is limited in practice, however, by such factors as the cost of the boiler and the friction introduced in the path of the flue gases. In small boilers it is usual to allow 1 sq. ft. of grate surface to every 15 to 30 sq. ft. of heating surface. For boilers of 50 hp. and over it is usual to allow from 30 to 40 sq. ft. of heating surface per square foot of grate surface, while in very large boilers the ratio is 50:1 or 60:1. Experience has shown that in small heating boilers it is advisable to allow each square foot of heating surface to evaporate only about 2 lb. of water per hr., as a greater rate of steaming results in a high exit temperature of the flue gases, and frequent firing is required. In large boilers the evaporation rate varies from 3 to 10 lb. per sq. ft. of surface.

Small heating boilers are distinctly different in operation from large power or heating boilers. In the latter, coal is being fed to the boiler almost continuously, and the flues are carrying a large quantity of gases. Small heating boilers, on the other hand, are fed with coal only at infrequent intervals, and less heat is transmitted to the water by the flue surfaces. The greater part of the heat is transmitted by the fire surfaces, that is, those in the paths of the heat rays emanating from the fuel bed. It is good practice to have about 60 per cent fire surface and 40 per cent flue surface in small cast-iron boilers for coal burning.

121. Boiler Rating.—The standard unit of boiler capacity is the boiler horsepower, which is defined as the equivalent of 34.5 lb. of steam evaporated per hour "from and at" 212° (that is, from water at 212° into saturated steam at the same temperature). As each pound of steam so evaporated requires the transmission of 970.3 B.t.u., the boiler horsepower is equivalent to 33,475 B.t.u. per hr. It is customary to allow 10 sq. ft. of heating surface per boiler horsepower for establishing the rated capacity of a boiler. On this basis, 1 sq. ft. of surface when working at rated capacity evaporates 3.45 lb. of water per hr. Under ordinary conditions of operation with feed-water temperatures considerably below 212°, 1 b.hp. is equivalent to roughly 30 lb. of steam per hr.

The term "boiler horsepower" is gradually passing out of use because the high rates of steaming now used in power boilers have made it rather meaningless. Heating boilers are not usually rated in horsepower but by the amount of radiation they will handle or in B.t.u. per hour.

122. Draft and Chimney Construction.—In order to maintain combustion in a furnace, a continuous supply of air must be moved through the fuel bed. In the ordinary heating boiler, the air is drawn through by means of a chimney, which also serves to dispose of the smoke and other products of combustion. The chimney produces a draft, or movement of the air, because of the difference in weight between the column of hot gases in the chimney and the cold outside air. The intensity of the force produced depends upon the average difference in temperature between the hot gases in the stack and the outside air and upon the height of the stack. This force must be sufficient to move the required amount of air and gases through the boiler and stack against the frictional resistances interposed by the various obstructions. These resistances consist of (*a*) the resistance of the fuel bed, (*b*) the resistance of the flues of the boiler, (*c*) the resistance of the damper and breeching, and (*d*) the resistance of the stack itself. The first three items are fixed by the kinds of fuel used and by the design of the boiler. The last item depends upon the height, cross section, and construction of the stack. If the cross-sectional area of the stack is too small, the friction in the stack itself will be great, and the sum of the various resist-

TABLE 38.—CHIMNEY SIZES*

Warm-air furnace capacity in sq. in. of leader pipe	Steam-boiler capacity sq. ft. of radiation	Hot-water heater capacity sq. ft. of radiation	Nominal dimensions of fire clay lining in inches	Rectangular flue		Round flue		Height above grate, ft.
				Actual inside dimensions of fire clay lining, inches	Actual area, sq. in.	Inside diameter of lining, inches	Actual area, sq. in.	
790	590	973	8½ × 13	7 × 11½	81			35
1,000	690	1,140				10	79	
	900	1,490	13 × 13	11¼ × 11¼	127			
	900	1,490	8½ × 13	6¾ × 16¼	110			
	1,100	1,820				12	113	40
	1,700	2,800	13 × 18	11¼ × 16¼	183			
	1,940	3,200				15	177	
	2,130	3,520	18 × 18	15¾ × 15¾	248			
	2,480	4,090	20 × 20	17¼ × 17¼	298			45
	3,150	5,200				18	254	50
	4,300	7,100				20	314	
	4,600	7,590	20 × 24	17 × 21	357			
	5,000	8,250	24 × 24	21 × 21	441			55
	5,570	9,190		24 × 24†	576			60
	5,580	9,200				22	380	
	6,980	11,500				24	452	65
	7,270	12,000		24 × 28†	672			
	8,700	14,400		28 × 28†	784			
	9,380	15,500				27	573	
	10,150	16,750		30 × 30†	900			
	10,470	17,250		28 × 32†	896			

* A.S.H. & V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings.

† Dimensions are for unlined rectangular flues.

ance factors may be greater than the available draft produced by the stack. Increasing the area of the stack results in a reduction of its frictional resistance and therefore in an increase in the net amount of draft available at the foot of the stack for overcoming the boiler and breeching losses. Increasing the height of the stack obviously increases the available draft.

The dimensions of a chimney can be computed from a consideration of the principles stated above,[1] but for ordinary cases they can be determined by empirical rules. Table 38 gives the dimensions of chimneys for various amounts of steam or water radiator surface.

The available draft of such chimneys, properly designed and constructed, as measured with an ordinary draft gage, should approximate the values given in Table 39.

In measuring the available draft the gage should be connected to the breeching on the chimney side of the damper. The fire should be regulated so that the temperature of the stack gases will approximate working conditions, and the damper should be quickly closed immediately before the reading is taken.

TABLE 39.—DRAFT IN SMALL CHIMNEYS[1]

Height in feet	Temperature of chimney gases, deg. F. 200	250	300
	Draft—inches of water		
60	0.27	0.32	0.35
55	0.25	0.29	0.32
50	0.23	0.26	0.29
45	0.21	0.23	0.26
40	0.18	0.21	0.23
35	0.16	0.19	0.20
30	0.14	0.16	0.17
25	0.12	0.14	0.14
20	0.09	0.11	0.12

[1] From HAROLD L. ALT, "Chimneys: Their Design and Construction," *Heating and Ventilating Mag.*, March, 1917.

A chimney must be so constructed that the wind deflected by surrounding buildings will not blow down into it and thus impede the draft. The chimney should extend well above the top of all adjacent buildings.

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[1] For methods of chimney design see J. G. Mingle, "Draft and Capacity of Chimneys."

The round flue is the most effective per square foot of area but is somewhat difficult to construct. For small buildings a square or retangular flue is used. It should be lined with tile and should be smooth and free from leaks. Offsets should always be avoided, if possible, and when unavoidable should be made with gradual bends. No other openings (except a cleanout door) should be made in the flue to which the boiler is connected.

In large buildings the stack is constructed of steel and is usually lined with brick or tile.

123. Hot-water Boilers.—The "boiler" or heater designed for heating systems using hot water is quite similar in general design to a steam boiler. The chief difference is the omission of the steam dome. For large buildings or groups of buildings, ordinary steel fire-tube or water-tube boilers are often used for hot-water systems, although in many cases the water is heated by the exhaust steam from engines or turbines in a tubular heater.

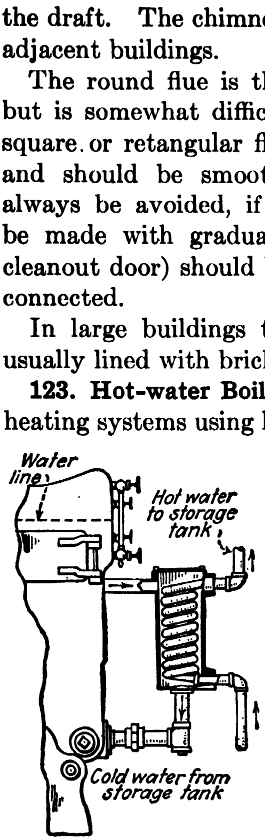

Fig. 79.—Water heater attached to a steam boiler.

124. Domestic Water Heaters.—The heating of water for lavatory and other domestic purposes may be done by the well-known method of installing a pipe coil in the fire pot of the furnace or boiler in connection with a storage tank. This is a simple and fairly satisfactory method but has the objection of obstructing the fire pot. Also, the supply of hot water is apt to be quite variable, depending upon the rate of combustion in the fire pot.

A better method with a steam boiler is to use a heater of the form shown in Fig. 79. Water from the boiler, at boiling temperature, circulates through the shell of the heater, while the water to be heated circulates through the coil and back to the storage tank by gravity. When a coil or water heater is to be installed, due allowance should be made for it in choosing the size of the boiler, as will be discussed later.

In large installations it is preferable to use steam for heating the water, by means of a tubular heater of the form shown in Fig. 80 or 81. The former is of the storage type with steam in the tubes and water in the surrounding space. This type is used where a storage of hot water is required because of intermittent use.

Where an "instantaneous" heater is desirable, as for heating a swimming pool or for use in connection with a separate tank, the form shown in Fig. 81 is used. The steam fills the shell, the condensate being drained off at the bottom, and the water is in the tubes.

In selecting the size of heater to be used it is usually satisfactory to depend upon the manufacturer's ratings.

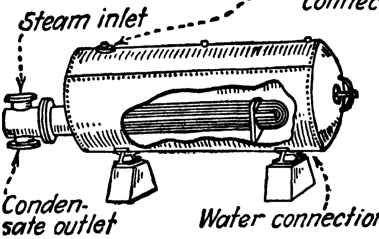

FIG. 80.—Tubular water heaters.

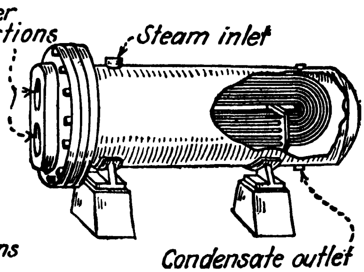

FIG. 81.

125. Determination of the Proper Size of Boiler.—Naturally the first step in selecting the proper size of boiler for heating a building is to compute the load in B.t.u. or in equivalent square feet of radiator surface that the boiler must handle.

The **design load** consists of the following items:

1. Heat required by all heat-emitting units with normal room temperatures and (in the case of heaters taking outdoor air) with the minimum outdoor temperature. This is frequently referred to as the **net load** in many testing and rating codes.
2. Heat emitted from all covered and uncovered steam piping.
3. Heat required by attached water heaters or other devices.

This is the load that must be provided for during the coldest weather *after* the building has been brought up to temperature. Since most buildings are allowed to cool down at night, the true maximum load will be considerably greater. The **maximum or gross load** consists of the design load *plus* the **starting load,** which includes

4. The additional heat emission of the radiators and piping, caused by the lower room temperatures existing during the warming-up period.

5. The heat required to raise the metal in the radiators and piping to the working temperature.

The direct-radiation load will have been expressed in terms of equivalent radiation. For direct-indirect radiators add 25 per cent to reduce to equivalent radiation, and for indirect radiation add 50 per cent. For fan heaters convert the estimated B.t.u. output to equivalent direct radiation.

When a water-heating coil is to be used in the firebox, heating water for a storage tank, take the storage-tank capacity in gallons and multiply by 2 for steam and by 3 for hot water to convert to equivalent square feet of radiation. When a water heater of the type through which the water from the boiler circulates is to be used, allow ½ sq. ft. of steam radiation for each gallon of tank capacity.

The heat emitted from the piping, both bare and covered, is an important part of the boiler load. It is customary to treat pipes covered with insulation less than ¾ in. thick as bare pipes. The heat emission from bare pipes may safely be taken as 2 B.t.u. per hr. per sq. ft. of pipe surface per degree difference between the steam (or water) and the surrounding air for steam pipes, and 1.8 B.t.u. for pipes in a hot-water system. When pipe covering ¾ in. or more in thickness is used, the heat emission should be taken as 25 per cent of the amount given above for bare pipes.

The maximum load is usually calculated by adding a percentage allowance to the design load to cover items 4 and 5. The amount of this allowance called for by the A.S.H. & V.E. Code is given in Table 40 for various design loads.

TABLE 40.—ALLOWANCE FOR STARTING LOAD

Design Load, B.t.u. per Hour	Percentage to Be Added for Starting Load
Up to 100,000	65
100,000 to 200,000	60
200,000 to 600,000	55
600,000 to 1,200,000	50
1,200,000 to 1,800,000	45
Above 1,800,000	40

As an example of this method, assume a steam heating system having 11,500 equivalent sq. ft. of radiator surface, 2,100 sq. ft.

of well-insulated pipe surface, and 300 sq. ft. of bare pipe surface. A water-heating coil in the boiler heats an 80-gal. tank. What are the design load and the maximum load?

$$\text{Design load} = 11{,}500 + \left(\frac{2{,}100}{4} + 300\right) \times (219.4^{*} - 70) \times \frac{2}{240} + (80 \times 2)$$

$$= 12{,}687 \text{ sq. ft., or } 3{,}044{,}880 \text{ B.t.u. per hr.}$$

$$\text{Maximum load} = 12{,}687 \times 1.40 = 17{,}762 \text{ sq. ft., or } 4{,}262{,}832, \text{ B.t.u. per hr.}$$

The boiler load having been calculated, it is possible to select a boiler of the proper size, as will be explained later.

126. Combustion Rates.—The output of any boiler is equal to the heat developed in the furnace multiplied by the boiler efficiency. The heat developed in the furnace depends upon the calorific value of the fuel and the rate of combustion. This relation may be expressed as follows:

$$Q = W \times G \times H \times E \tag{4}$$

where Q = boiler output, B.t.u. per hr.
W = weight of fuel burned per hr. per sq. ft. of grate area.
G = grate area, sq. ft.
H = calorific value of fuel, B.t.u. per lb.
E = combined efficiency of boiler and grate.

The proper rate of combustion per square foot of grate area is fixed by certain practical considerations. Heating boilers are ordinarily charged with fuel at periodic intervals, and the furnace must be large enough to hold sufficient fuel for several hours' operation plus enough to kindle the succeeding charge. The rate of combustion per square foot of grate area must therefore be low enough to give a reasonably long period between firings. Too high a combustion rate may also cause trouble from clinkers. Higher combustion rates are used with large boilers than with small boilers, partly because it is assumed that more frequent attention can be given to the fire. Table 41 gives the maximum combustion rates suitable for various fuels and boiler sizes.

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* Temperature of steam at 2.3 lb. gage.

TABLE 41.—MAXIMUM COMBUSTION RATES[1]
(Pounds fuel per hour, per square foot of grate)

Grate area, sq. ft.	Fuel	Maximum combustion rate
Up to 4	No. 1 buckwheat anthracite	3
5–25	No. 1 buckwheat anthracite	3½–5
Up to 9	Anthracite pea	5
10–25	Anthracite pea	5½–6
Up to 4	Anthracite nut and larger	8
5–25	Anthracite nut and larger	9–13
Up to 4	Bituminous	9½
5–14	Bituminous	12
15 and above	Bituminous	15.5

[1] Abstracted from A.S.H. & V.E. "Guide," 1945.

The maximum combustion rate that can be obtained in any boiler installation is limited by the available chimney draft. The chimney must be large enough and tall enough to pull the requisite amount of air through the fuel bed.

The boiler efficiency directly affects the boiler output. It depends upon the amount of heating surface in the boiler, the arrangement of the heating surface with reference to the flow of gas, the cleanliness of the surface, and the combustion rate.

127. Selecting a Boiler.—In practice there are three methods used to select the proper size of *cast-iron* boiler for a given load. (Steel boilers are usually rated according to special rules recommended by the Steel Boiler Institute, which will be discussed later.) They are

1. The use of the boiler manufacturer's ratings.
2. Selection on the basis of grate area and heating surface.
3. The use of actual output tests.

If the manufacturer's ratings are known to be conservative and reliable, and if their exact basis is known, there is no objection to placing dependence upon them. However, much confusion has arisen because of the widely different margins that different manufacturers allow in their published ratings to cover piping loss, starting load, etc. In some cases this allowance is made in fixing the rated capacities, and the boilers are capable of supplying the amount of actual radiation for which they are rated. Other manufacturers do not include such allowances.

The second method is recommended by the Heating, Piping and Air Conditioning Contractors National Association and is based on the physical characteristics of a boiler. A boiler of a given grate area is assumed to be capable of supplying a given amount of equivalent radiation, including the normal piping loss, starting load, etc. In round boilers where the heating surface may be varied by adding intermediate sections, credit is allowed for increased heating surface, and the ratings are increased. Ratings are expressed on a **net load** basis and have sufficient excess capacity to supply the normal piping and starting load.

The advantage of the Heating, Piping and Air Conditioning Contractors National Association's method is its simplicity. It assumes, however, that the performance of all makes of boilers will bear approximately the same relation to their physical dimensions. This is probably true of boilers of conventional design but might not hold for untried or freak designs.[1]

TABLE 42.—EXAMPLE OF METHOD OF RATING BOILERS RECOMMENDED BY THE A.S.H. & V.E., 1930

(Boiler designation, S-O-4; grate area, 11.94 sq. ft.; available fuel-holding capacity, 627 lb.)

Output, B.t.u. per hour	1,255,000	864,000	700,000	563,000	466,000
Output eq. steam rad., sq. ft.	5,220	3,600	2,920	2,355	1,940
Output eq. water rad., sq. ft.	8,367	5,760	4,667	3,754	3,106
Fuel available, hr	4	**6.4**	8	10	12
Rate of combustion, lb. per sq. ft. per hour	13.12	**8.5**	6.56	5.25	4.37
Efficiency, per cent	64.0	**69.6**	71.5	72.0	71.5
Average draft, in. water	0.14	**0.12**	0.06	0.03	0.02
Average flue-gas temperature, °F.	540	**502**	460	420	380
Inside dimens. chimney, in.	16 × 20	**16 × 16**	16 × 16	12 × 16	12 × 12
Min. height of chimney, ft.	55	**50**	45	45	45

The third method of selecting a boiler is by the use of actual output tests as recommended by the Codes of the A.S.H. & V.E. and the I.B.R. The A.S.H. & V.E. Code for Rating Steam Heating Boilers (adopted 1929, revised 1930) assumes that the manufacturer will conduct laboratory tests according to the Code by determining the performance at five different outputs ranging from the maximum output (established by the manufacturer) down to 30 per cent of the maximum. The design load (Art. 125) is to be one of the outputs for which performance data are given and is printed in heavy type. An example of the method suggested in the Code for the presentation of such data by the manufacturer is given in Table 42, and Fig. 82 shows the performance curves plotted from the tabulated data.

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[1] Tables giving the net radiation loads for round and sectional cast-iron boilers have been compiled by the association in the form of Engineering Standards, Part II, 1943, and Net Load Recommendations for Heating Boilers, December, 1943.

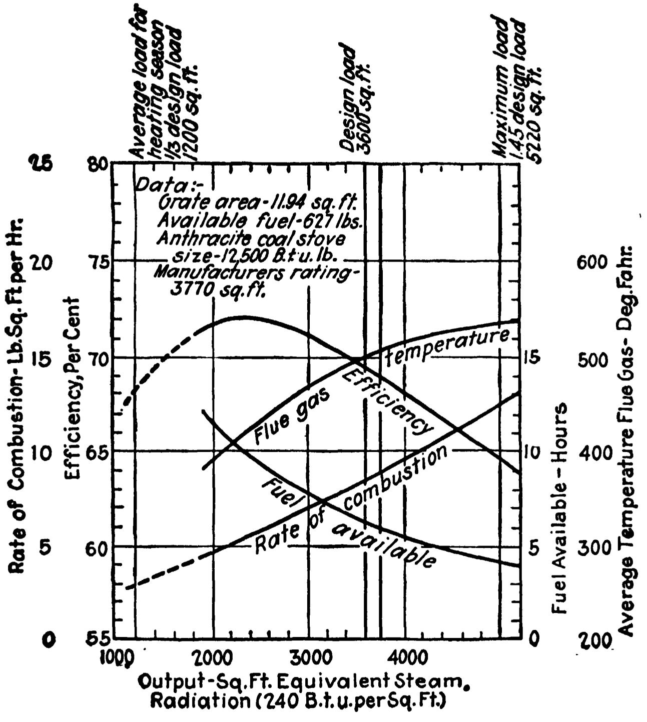

FIG. 82.—Boiler-performance curves.

The advantage of the A.S.H. & V.E.'s method is that it gives a fairly complete picture of the performance of the boiler under the calculated load conditions. It should be borne in mind, however, that the efficiency figures are test figures and will not ordinarily be equaled in actual operation.

Figure 82 illustrates the typical variation of the boiler performance with increasing output. The increase in flue-gas temperature and the decrease in efficiency as the output increases are clearly shown.

These particular performance figures are based on the use of anthracite coal. If bituminous coal or coke is to be used, it is desirable to obtain similar figures based on these fuels. The "fuel available" will be less than with anthracite, because of the greater space occupied per pound of fuel.

This method of rating boilers is very good in theory, but it does not give a truly standard method of rating. The manufacturer recommends the size of boiler for the job or, in other words, fixes the position of the heavy vertical line in Fig. 82, and each manufacturer may therefore offer a different size of boiler for the same job. A boiler may be capable of handling the load required but may do so at a low efficiency owing to a shortage of heating surface. Efficiency as well as capacity is important in choosing a boiler.

A weakness of the A.S.H. & V.E. Code is the lack of standardization for determining efficiency or flue-gas temperature at the design load. The I.B.R. Testing and Rating Code for Low Pressure Heating Boilers (1945) overcomes this objection for rating cast-iron sectional boilers by providing standardized procedures for determining the gross output based on an evaporation test.

The I.B.R. Code applies to the testing of hand-fired and oil-fired boilers, regardless of size. The limiting flue-gas temperature in the oil-fired test is 600° with the CO_2 set at 10 per cent, and the over-all efficiency for establishing the gross output or maximum load is to be not less than 68 per cent of the heat input. The net radiation load may be established from tables that provide the necessary allowances for piping and pickup or starting-load factors.

128. Selection of a Steel Boiler.—The method of choosing the proper size of a steel boiler is somewhat different from that of choosing a cast-iron boiler, not so much because it is made of a different material as because of differences in design and in trade customs. The Steel Boiler Institute, a manufacturers' organization, has established a code[1] for the rating of steel boilers. Catalogue **S.B.I. ratings** established in accordance witht his Code are intended to correspond with the estimated **design load** (direct radiation plus hot-water load plus heat loss from piping).

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[1] The following material is based on the Rating Code as revised June 12, 1945.

For boilers in which hand-fired solid fuel is burned, the **rating** of a steel boiler expressed in square feet of steam radiator surface is **14 times the heating surface** of the boiler in square feet. One square foot of steam radiator surface is considered to have an emission of 240 B.t.u. per hr., and 1 sq. ft. of hot-water radiator surface is considered to have an emission of 150 B.t.u. per hr.

For boilers in which mechanically fired solid fuel or oil or gas is burned, the rating in square feet is **17 times the heating surface** of the boiler in square feet.

The **grate area** of a boiler for the rating thus determined shall be not less than that determined by the following formulas:

For boilers with ratings 1,800 to 4,000 sq. ft. of steam radiation

$$\text{Grate area} = \sqrt{\frac{\text{catalogue rating (in sq. ft. steam radiation)} - 200}{25.5}} \qquad (5)$$

For boilers with ratings of 4,000 sq. ft. of steam radiation and larger

$$\text{Grate area} = \sqrt{\frac{\text{catalogue rating (in sq. ft. steam radiation)} - 1{,}500}{16.8}} \qquad (6)$$

Heating surface is considered as including all surfaces that are exposed to the products of combustion on one side and water on the other side. The outer surface of tubes is used. Grate area in the case of double-grate boilers is taken as the area of the upper grate plus one-fourth of the area of the lower grate.

The furnace volume for furnaces in which solid fuel is burned shall be considered as the cubical content of the space between the bottom of the fuel bed and the first plane of entry into or between the tubes. Furnace volume for furnaces in which pulverized fuel or liquid or gaseous fuel is burned shall be considered as the cubical content of the space between the hearth and the first plane of entry into or between the tubes. No minimum furnace volume shall be specified for mechanically fired boilers burning anthracite.

The **furnace volume** for a boiler in which pulverized fuel, oil, gas, or stoker-fired bituminous coal is burned shall be not less than 1 cu. ft. for every 140 sq. ft. of steam rating.

The *average* height of furnace in which stoker-fired bituminous coal is burned shall be not less than that determined by the following formula:

$$H = \sqrt{\frac{R}{22.5}} + \sqrt{\frac{R}{A}} \tag{7}$$

where H = average furnace height, in., as determined by the following formula:

$$H = \frac{12F}{A} = \frac{12F}{WL} \tag{8}$$

R = stoker-fired boiler rating, sq. ft. steam radiation.
A = plan area of firebox, sq. ft., measured at the bottom of the fuel bed.
F = furnace volume, cu. ft.
W = average width of furnace, ft., measured at the bottom of the fuel bed.
L = length of furnace, ft. If the furnace is longer than the fuel bed or contains a bridge wall, the total length of the furnace may be used, except that this length shall not exceed $2\frac{1}{2}W$.

The foregoing empirical formulas are so designed that the combustion rate per square foot of grate area varies from about 5 lb. per hr. for a boiler rated at 500 sq. ft. of radiation to about 18 lb. per hr. for a boiler rated at 20,000 sq. ft. of radiation.

The latest edition of the Code[1] specifies, in tabular form, *minimum* furnace heights for commercial steel boilers stoker-fired with bituminous coal.

Example.—Determine the required heating surface and grate area of a steel boiler for a building whose design load is 1,850 sq. ft., assuming (*a*) that hand-fired solid fuel will be used. Also determine the required heating surface and furnace volume if (*b*) an oil burner is used.

a. Heating surface, 1,850 ÷ 14 = 132 sq. ft.

$$\text{Grate area, } \sqrt{\frac{1{,}850 - 200}{25.5}} = 8 \text{ sq. ft.}$$

b. Heating surface, 1,850 ÷ 17 = 109 sq. ft.
Furnace volume, 1,850 ÷ 140 = 13.2 cu. ft.

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[1] Rating Code, Steel Boiler Institute, June 12, 1945.

A boiler having not less than the required heating surface, grate area, or furnace volume would be chosen from the manufacturer's catalogue for either case. The steel-boiler manufacturers usually publish the ratings for their boilers determined on the foregoing basis.

The S.B.I. Code also includes a testing procedure for establishing ratings of hand-fired and automatically fired residential steel boilers containing not more than 177 sq. ft. of heating surface. The Code prescribes that the **S.B.I. net rating** (direct radiation plus hot-water load), expressed in square feet of steam radiation, shall not be greater than 17 times the square feet of heating surface for oil-fired boilers and 14 times the heating surface for hand-fired units.

The tests for oil-fired boilers stipulate that the flue-gas temperature shall not exceed 600° when the boiler is operating at 150 per cent of **S.B.I. net rating** with a CO_2 of 10 per cent and 70 per cent over-all efficiency. The gas- and stoker-fired ratings shall be not greater than the oil-fired boiler ratings.

129. Sizing Boilers for Oil and Gas.—Present-day commercial practice in selecting boilers is not entirely satisfactory. Competition often impels the manufacturer to recommend a size of boiler purely from the standpoint of its load-carrying ability and without regard for economy, in the effort to reduce the cost of the boiler. Such a practice may sometimes be justified, but it should be realized that a boiler may be capable of carrying the required load yet at the same time be very inefficient.

Particularly is this true of boilers in which oil or gas is to be burned. With solid fuels the boiler is operated at its maximum load only a very few hours during the year, and at other times the amount of heat developed in the combustion chamber is small compared to the area of the heat-absorbing surfaces in the boiler, and the gaseous products of combustion therefore leave the boiler at a low temperature. With the oil burner, however, and usually with the gas burner, the control operates on the intermittent principle, and the burner is either working at its maximum capacity or shut off entirely. Consequently the amount of heat-absorbing surface, especially in the form of convection surface, should be more generous than if solid fuel is burned. With gas this is even more important because of the higher cost of gas, and a well-designed gas-fired boiler has a liberal amount of convection surface.

The same principle applies, to a degree, when a stoker is to be used, because most small stokers operate intermittently and when operating have a high combustion rate.

All of this emphasizes the desirability of choosing boiler sizes from actual tests as recommended by the A.S.H. & V.E. or the I.B.R., because the designer then knows fairly well how the boiler will actually perform.

The volume of the combustion chamber in a boiler in which oil is to be burned should be at least 1 cu. ft. for each 1.5 to 2 lb. of oil burned per hr.

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Problems

1. What is the cost of heating per 1,000,000 B.t.u. delivered to the heating system for the following fuels?

Anthracite coal at $16 per ton, 13,700 B.t.u. per lb.
Coke at $12 per ton, 14,000 B.t.u. per lb.
Oil at $0.11 per gal., 138,500 B.t.u. per gal.
Gas at $0.80 per 1,000 cu. ft., 550 B.t.u. per cu. ft.

Assume efficiency of boiler and furnace to be 55 per cent for coal and coke, 70 per cent for oil, and 75 per cent for gas.

2. A steam heating system has 15,000 sq. ft. of equivalent radiator surface, 2,500 sq. ft. of pipe surface well insulated, and 400 sq. ft. of bare pipe surface. A water heater is attached to the boiler, through which the boiler water circulates, and the capacity of the water-storage tank is 100 gal. What is the design load? the maximum load?

3. A boiler having a grate area of 18 sq. ft. burns 9 lb. of coal per sq. ft. per hr. with a boiler and grate efficiency of 70 per cent. The coal has a heating value of 13,000 B.t.u. per lb. What is the boiler output in B.t.u. per hour?

4. A stoker is to burn 350 lb. of bituminous coal per hr. under a steel boiler. The grate area is 36 sq. ft. What is the preferred furnace height? the heat release?

5. What should be the coal-burning capacity of a stoker for a boiler supplying 7,400 sq. ft. of steam radiator surface and a water heater supplying 100 gal. per hour of water at 130°? Bituminous coal; city water temperature 40°.

6. What should be the grate area of a stoker-fired steel boiler rated at 4,750 sq. ft. of steam radiation? What should be the furnace height and the heating surface?

CHAPTER VII

STEAM HEATING SYSTEMS

130. Single-pipe Systems.—In a steam heating system the piping and radiators must be arranged with a view to performing successfully three functions: (1) the conveying of steam to the radiators, (2) the removal of air, and (3) the draining off of the condensation. The many types of steam heating systems in use differ from one another mainly in the manner in which these operations are accomplished.

Steam heating systems may be divided roughly into two general classes, according to the manner in which the connections are made to the radiators. In **single-pipe** systems the steam is conveyed to the radiator through a pipe that enters the radiator at the bottom of one of the end sections. The condensation that forms in the radiator flows back through this same pipe. In **two-pipe** systems a separate system of piping is provided to carry away the condensation, and in some cases also the air, from the radiators.

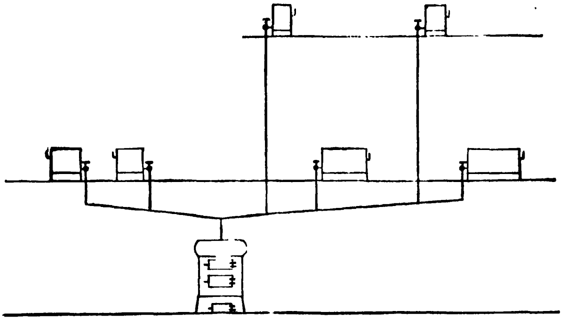


FIG. 83.—Single-pipe system—mains pitching toward boiler.

The simplest form of single-pipe system is that shown diagrammatically in Fig. 83. The nearly horizontal pipes leaving the boiler are called the steam **mains.** The vertical pipes extending to the upper floors are called **risers.** Steam is generated in the boiler and flows through the mains and risers into the radiators, forcing the air out ahead of it through some kind of air valve on the end of the radiator opposite the supply connection. In the system shown in Fig. 83 the condensation formed in the radiators drains down the risers into the mains and back to the boiler. The direction of the flow of the condensation is thus opposite to the direction of the steam flow.

In the risers this is not objectionable if the system is small, but in the mains the water and steam flowing in opposite directions are very liable to interfere with each other, unless the pipes are of such a diameter that the steam will travel at a very low velocity. If the pipes are small so that such interference takes place, the water is picked up by the steam and driven to the end of the main with a characteristic loud cracking noise known as "water hammer."

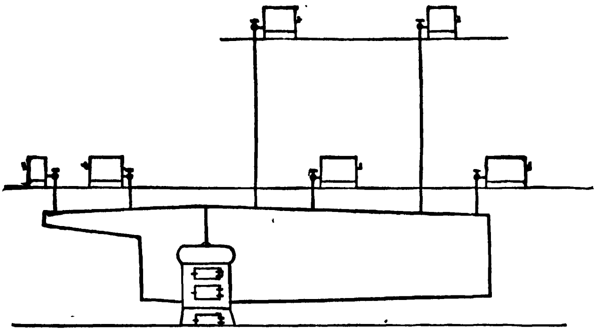


FIG. 84.—Single-pipe system—mains pitching away from boiler.

A better design of a single-pipe system is shown in Fig. 84. The main pitches away from the boiler, and the condensation entering the main from the risers flows along with the steam. The main circles the basement, and a drip connection carries the condensation from the end of it to the boiler, entering below the water line. This is the most common form of single-pipe system. (For the sake of simplicity these heating systems are here shown in their elementary form. The actual pipes would be installed somewhat differently, as will be shown later.)

The single-pipe system is simple in design and can be installed at a low cost. It is especially suitable for residences and small buildings where a low-priced system is desired. In large buildings a single-pipe system is less desirable, on account of the large quantities of water that must be carried in the steam mains and risers. Another objection is the trouble sometimes experienced by radiators' not draining properly. If the inlet valve is not closed tightly when the radiator is shut off, or if the valve leaks, some steam will continue to flow into the radiator, and because of the small area of the opening it is impossible for the condensation to drain out against the inflowing steam. As a result the radiator becomes partly filled with water; and when the valve is again opened, an annoying cracking and pounding take place as the water pours out against the inrushing steam.

131. Two-pipe Systems.—Figure 85 shows a typical two-pipe dry-return system. As the term indicates, the return mains are above the water line of the boiler and are filled with steam.

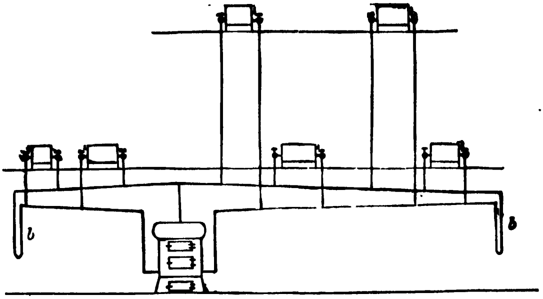


FIG. 85.—Two-pipe dry-return system.

The supply mains and risers are installed and connections are taken from them to each radiator in much the same manner as in the single-pipe system. A return connection is made from each radiator to the return main, through which the condensation from the radiator flows. As the steam has a free passage through the radiator from the supply main to the return main, it is evident that the latter will be filled with steam at a pressure approaching that in the supply mains, a slight pressure drop taking place through the radiator and its connections. The end of each supply main is dripped into the return main through a 4- or 5-ft. water seal as at *b*, *b*, which serves to prevent the full steam pressure from entering the return main. One of the chief faults of the two-pipe dry-return system is the tendency of the steam to enter the radiator through the return connection, especially if the return valve is opened first when the radiator is being turned on, thus trapping air in the center of the radiator. This type of system is obsolescent.

In the "wet-return" system the return main is below the water line of the boiler, and separate connections are made to it from each radiator and from the low points in the supply mains. In the wet-return system shown in Fig. 86 it is evident that no steam can enter the radiator through the return connection, as the lower end of each connection is sealed with water. It is also evident that the water level in the return pipes is sometimes considerably higher than in the boiler. If the pressure on the surface of the water in the boiler is the same as that on the surface of the water in the return lines, then the water levels will be the same. But if a pressure of 2 lb., for example, exists in the boiler, and

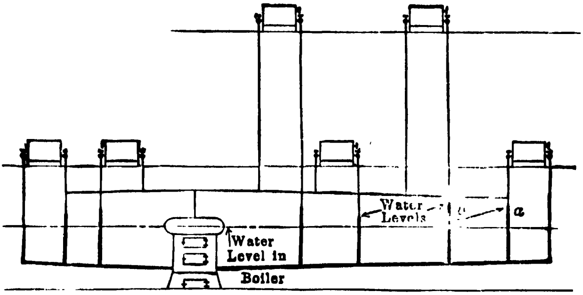


Fig. 86. -Wet-return system.

there is a drop, due to friction, of ½ lb. along the main, then the water at *b* will rise to a height sufficient to balance the drop between the boiler and the point *b*. It is necessary, therefore, to use pipes large enough to prevent an excessive pressure drop; moreover, no radiators should be located less than 2 ft. above the water line of the boiler. The wet-return system will usually operate with less noise than a dry-return system, as the condensation does not flow in horizontal pipes containing steam. A disadvantage of two-pipe systems is the cost of a double set of radiator valves and the nuisance of having to operate both valves. Sometimes a check valve is used instead of a shutoff valve on the return end of the radiator, but this arrangement is liable to give more or less annoyance from noise.

132. Overhead System.—In buildings over three or four stories high the overhead system illustrated in Fig. 87 is nearly always used. The main circles the attic, and risers extend downward from it to the basement, supplying the radiators on the successive floors. The steam is carried to the attic main by a main riser from which no radiators are supplied. The chief advantage of the overhead system of distribution lies in the fact that the steam and condensation in the risers are both moving downward. Smaller risers can therefore be used without causing noise or interfering with the circulation of the system. The fact that the large piping is in the attic rather than the basement is also an advantage when the matter of headroom and appearance in the basement is a consideration.

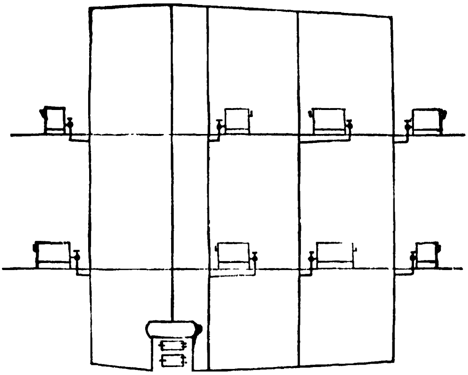


FIG. 87.—Overhead distribution—single-pipe system.

The overhead method of distribution may be applied to either the single-pipe or the two-pipe system. In the latter the return risers and the return main are arranged in the same manner as in an ordinary upfeed system.

133. Vapor Systems.—A modern form of two-pipe system with many desirable features is the so-called **vapor** system, which with slight modifications is sold under various trade names. The vapor system is so named because the steam circulates at a very low pressure—a few ounces above atmospheric. In the ordinary steam system the air must be forced out of the radiator through the very small orifice of the air valve, which requires a considerable pressure; but in the vapor system the path for the removal of the air is ample and unrestricted, and a pressure of a very few ounces is sufficient.

The arrangement of the radiator in a vapor system is shown in Fig. 88. The steam flows along the top of the radiator and downward into the various sections, driving the air out through the return pipe. The steam may fill only part of the volume of the radiator, as shown, if the amount supplied by the boiler is limited; or the supply can be regulated by partially closing the inlet valve.

At the point of discharge from the radiator is located the **trap,** whose function is to permit the air and condensation to pass into the return pipe but to prevent the steam from doing so.

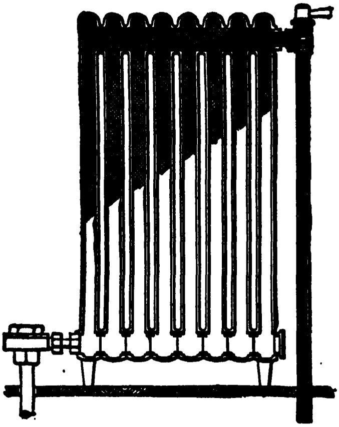


FIG. 88.—Radiator in a vapor system.

The surface of that part of the radiator which is filled with steam is at nearly the steam temperature. The remainder of the surface is warmed by the condensation that trickles down the inside surfaces. The temperature of the discharged condensation is thus materially lowered, and where the condensation is not returned to the boilers this is an advantage from the standpoint of economy.

One advantage often claimed for the vapor system is that the heat output of each radiator can be controlled by adjusting the opening of the inlet valve and thus partially filling the radiator. Although it is possible to do this, experience has shown that very few people will take the trouble to make such adjustments.

The low pressure at which the steam circulates is, however, of some advantage. The radiators become warm quickly, and the heating effect is more even and continuous than with the older type of steam system. This is particularly true in a small system. The low pressure also tends to more economical operation, although the degree of saving is somewhat uncertain. The ease of control, afforded by the location of the inlet valve at the top rather than at the bottom of the radiator and by the valves' being designed to open with a fraction of a turn, are other good features.

Summing up, the advantages of the vapor system are (1) improved method of air removal and eliminating noise, drip, and, smell; (2) ease of control; (3) somewhat more even heating; and (4) possible economy of operation.

The cost of the special fittings and appliances renders the vapor system more costly than the simpler types of steam systems, and somewhat more careful maintenance is required to keep the system in proper working order.

134. Inlet Valves.—An important part of the vapor system is the improved supply valve. Many designs have been developed, one of which is shown in Fig. 89*a*. This valve is of the packless type, having a flexible bellows that prevents leakage around the stem while permitting free movement of the stem and disk. The term "packless" is somewhat loosely used as applied to radiator valves. It covers, in general, all valves that do not require soft packing around the stem as in the common radiator valve. There are many designs using various kinds of hard-rubber and metallic packings, which are classed, with the bellows type, as packless. Figure 89*b* shows a valve of this kind, in which a rubber packing ring is compressed by a spring.

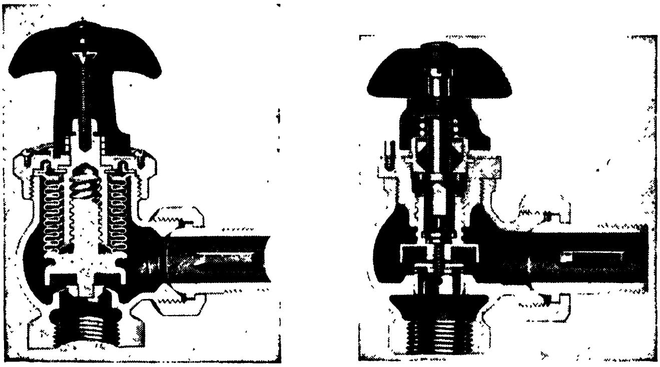

(*a*) (*b*)

FIG. 89.—Radiator inlet valves. (*Warren Webster and Company.*)

Another feature common to all supply valves of vapor systems is that they can be conveniently opened with less than one turn of the handle.

135. Traps.—The essential feature of the vapor system is the radiator trap. The successful operation of the system depends very largely upon it.

The most common type is the **thermostatic** trap, several commercial forms of which are shown in Fig. 90. It consists of a thin-walled chamber or bellows *A*, which contains a volatile liquid such as alcohol or kerosene. The liquid tends to volatilize with increasing temperature, forming sufficient pressure inside the bellows, at a temperature somewhat below 210°, to expand it and bring the valve *B* against the seat *C*.

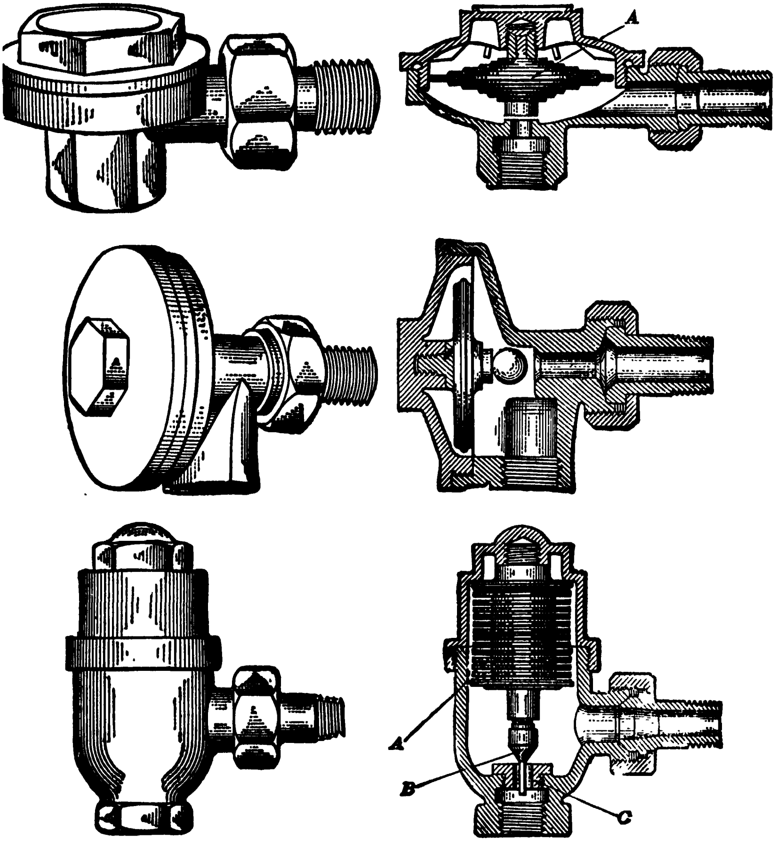


Fig. 90.—Various forms of thermostatic traps.

The movement of the bellows is governed by the resultant of the forces acting upon it. There is the vapor pressure of the liquid inside, which depends upon and varies with the temperature, and the pressures of the vapor and air outside, which are opposed to the internal pressure. Then there is the force exerted by the metal in the bellows, tending to preserve its original shape.

In actual operation the trap remains open sufficiently to allow the air and condensation to pass through it, but when steam reaches it and heats the thermostatic element it closes and remains closed until the condensation accumulating in it cools a few degrees, causing it to open again slightly.

Another type of radiator trap is the **float** trap, in which the opening and closing of the valve are dependent entirely upon the flow of condensation into the trap. The chief objection to float traps is that they are sometimes noisy in operation and are then a source of annoyance to the occupants of the room. There is also a tendency toward some leakage of steam through the trap.

The trap may be omitted entirely if the supply of steam is restricted in some manner so that no more steam can enter any radiator than is condensed in it and none can enter the return pipes. The restriction usually consists of an orifice disk placed in the radiator inlet.

Various devices to take the place of the radiator trap have been brought out from time to time but have not been widely used, and it may be said to be standard in vapor and vacuum heating systems.

136. General Arrangement of Vapor Systems.—The simplest arrangement of a vapor heating system is shown diagrammatically in Fig. 91. The return pipes, which in a vapor system carry only air and water, pitch back toward the boiler, and some sort of air valve is used to permit the egress of air but to prevent the escape of steam or water that might accidentally find its way to that point.

Some vent valves include also a check-valve arrangement that allows air to escape from the system but prevents it from reentering. The air is driven out of the system when the radiators and piping fill with steam; and as the steam output of the boiler decreases, the pressure falls below atmosphere, and the boiler continues to generate steam after the temperature of the water in it has dropped below 212°, as is the case in a vacuum system.

The elementary arrangement of Fig. 91 is fairly satisfactory but sometimes gives trouble as a result of the unbalanced pressure conditions. This will be understood by referring to Fig. 92. The vapor system normally has no pressure in the return pipes.

As the boiler pressure increases, an equivalent head of water builds up in the vertical return leg, and the pressure must not exceed that equivalent to the distance *h*, else the return main will become flooded, the system will not operate satisfactorily, and the water will not reenter the boiler. The building up of a few pounds pressure is difficult to prevent even with an automatic damper regulator because of the inertia effect of the fuel bed. A boiler designed with a low water line helps this situation somewhat, but the boiler pressure must always be very low.

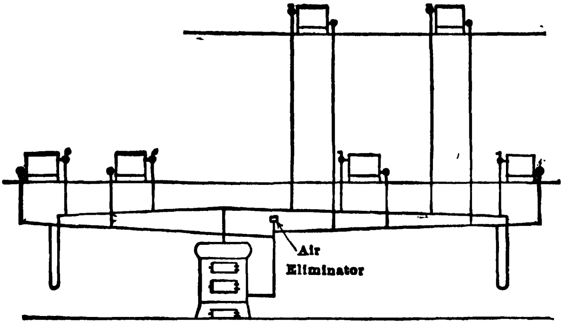


FIG. 91.—Arrangement of vapor system.

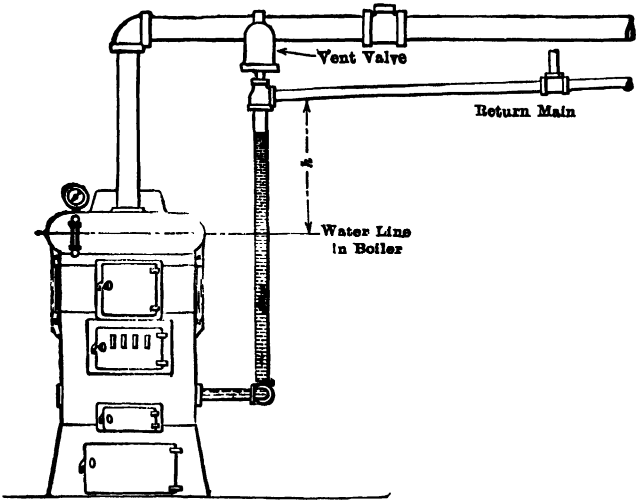


FIG. 92.

To ensure that a vapor system works properly in spite of occasional higher boiler pressure it is necessary to provide some means of forcing the water back into the boiler. This may be done in one of three ways: (*a*) by using a boiler feed pump, (*b*) by means of an alternating trap, or (*c*) by a differential seal.

A boiler feed pump is a satisfactory method on large systems and is frequently used. It requires a certain amount of attention, however, and therefore is not desirable in some buildings.

137. The Alternating Return Trap.—The alternating trap intermittently balances the pressures so as to allow the water to flow by gravity into the boiler. A trap of this sort is shown in Fig. 93, and the method of installation in Fig. 95.

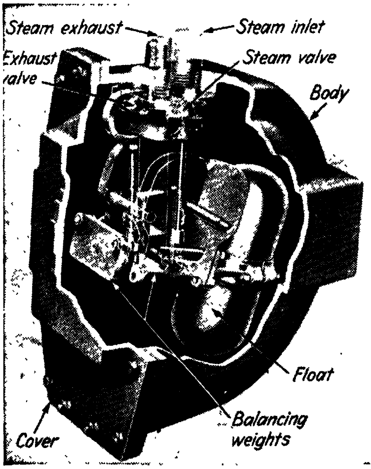


FIG. 93.—Boiler return trap. (*Warren Webster and Company.*)

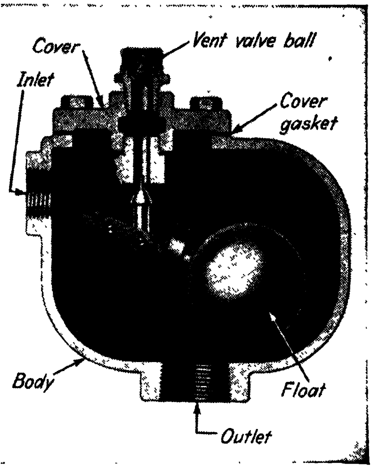


FIG. 94.—Vent trap. (*Warren Webster and Company.*)

The alternating trap is vented to the dry return and then through the vent trap shown in Fig. 94 to the atmosphere. When the boiler pressure is quite low, the condensation returns to the boiler directly; but when the pressure exceeds about 1 lb., the water is held back until it fills the alternating trap, the check valve *A* (Fig. 95) being closed. When the water level in the trap reaches a certain point, the float quickly closes the vent valve and opens a valve in the pipe leading from the boiler. This allows boiler pressure to build up in the trap above the surface of the water, and the water begins to flow into the boiler by gravity, closing the check valve *B*. When the trap becomes nearly empty, the valve admitting steam from the boiler is closed and that on the vent line is opened by the float, permitting the trap to refill. The cycle is then repeated. The alternating trap is a successful device and is rather widely used.

138. Vacuum Return-line Systems.—The return pipes in a vapor system carry the air and condensation back toward the boiler, the flow being caused by the pressure of the steam and by gravity.

In a large building it is difficult to accomplish this satisfactorily with gravity flow, and recourse must be had to the use of a suction pump, which maintains a pressure below atmosphere in the return piping and creates a positive flow of the air and condensation. Rapid filling of all the radiators with steam is assured because of the greater pressure differential through the system.

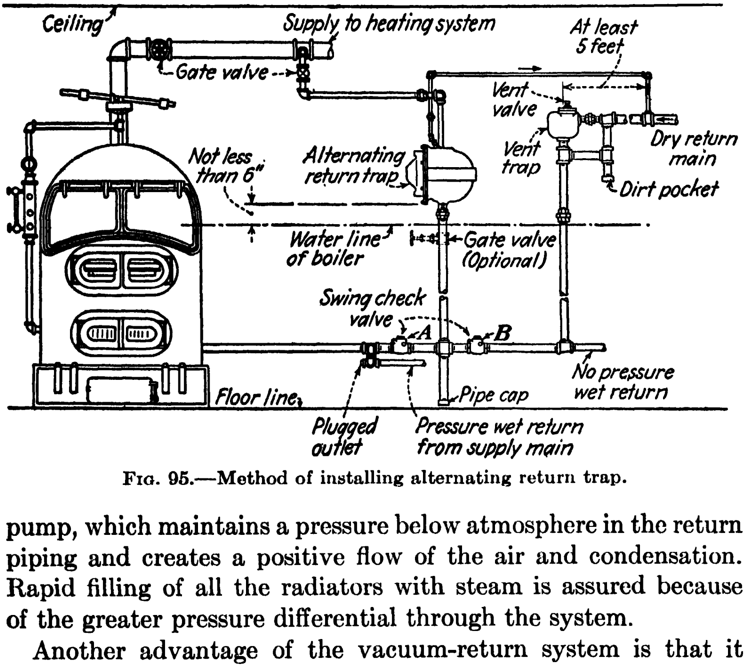


FIG. 95.—Method of installing alternating return trap.

Another advantage of the vacuum-return system is that it permits radiators to be located below the level of the return main or below the water level in the boiler.

A certain degree of control of the heat output of the radiators is possible with a vacuum system. The temperature of the steam, and consequently the heat emission of the radiator (Chap. V), varies with the pressure. If a pressure below atmosphere is maintained in the *radiators* by cutting down the supply of steam and operating the vacuum pump, the steam temperature will be reduced correspondingly, and the heat emission will be considerably lowered. In large buildings where there is no automatic temperature control, this method affords a means of reducing the steam consumption in moderate weather. When exhaust steam is used for heating, a vacuum system permits a lower back pressure on the engines and turbines and therefore improves the economy of the plant. Vacuum systems are best suited to large buildings where the advantages to be gained will justify their initial cost and operating cost.

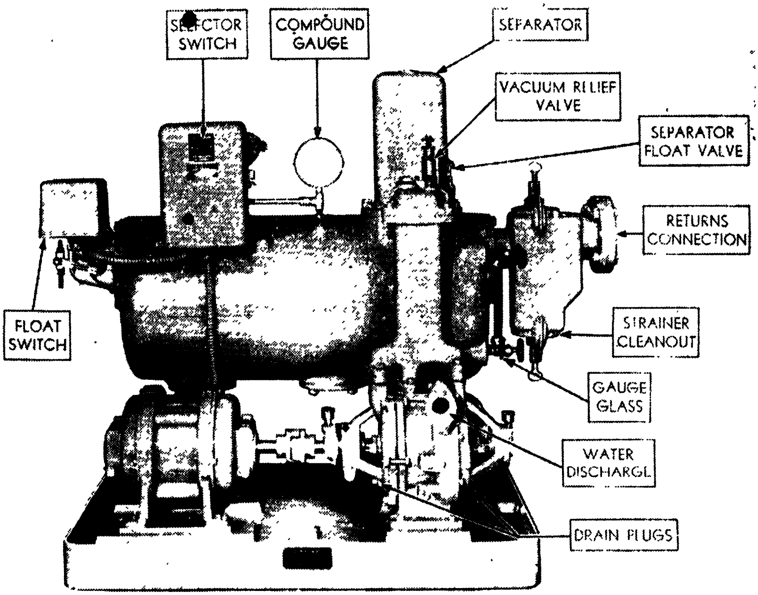


Fig. 96.—Motor-driven vacuum pump and receiver. (*Nash Engineering Company.*)

The vacuum system is arranged quite like the vapor system (Fig. 91) except that the return pipes are brought to the vacuum pump or its receiver.

139. Vacuum Pumps.—The vacuum pump in a vacuum-return system must handle both air and water. A motor-driven vacuum pump is shown in Fig. 96. The condensation flows into the reciever and is taken by the pump and returned to the boiler. In this design there are two impellers mounted on the same shaft, one handling the water and the other the air. An automatic control, actuated by the subatmospheric pressure in the receiver and by the water level, functions to start and stop the pump as required. A vacuum equivalent to 10 to 18 in. Hg can be easily maintained if the system is reasonably free from leaks.

For the proper operation of a vacuum system it is essential that the traps on the radiators be in good condition and close tightly. If they do not close tightly, a leakage of steam into the return pipes will occur, which will make it very difficult to maintain the vacuum.

140. Capacity of Vacuum Pumps.—Some types of rotary vacuum pumps have a separate element to handle the air. The air capacities in such cases may be specified as in Table 43.

The water capacity of the pump should be not less than ½ to ⅔ lb. of water per hour per square foot of radiation. Pumps handling both air and water should be capable of handling both the foregoing water quantity and the air quantity taken from Table 43 at a vacuum of at least 8 in. of mercury.

TABLE 43

Square Feet Equivalent Direct Radiator Surface	Air Capacity Cu. Ft. per Minute
8,000	5
16,000	9
26,000	15
40,000	19
65,000	34
100,000	60
150,000	80
300,000	180

The air must be separated from the water in some manner before the latter is fed to the boiler. In the pump of Fig. 96 this is accomplished in the receiver *before* the water goes to the water element of the pump, and the latter may therefore be used as the boiler-feed pump if the boiler pressure is low. For higher pressures or where reciprocating pumps are used, it is necessary to pump through an intermediate vented receiver tank. The receiver should be large enough to take care of a quantity of water, between its normal high and low levels, equal to 3 to 5 min. return at the maximum rate.

141. Boiler-feed Pumps.—For boiler feeding alone, the motor-driven centrifugal pump is used to good advantage, although in the small sizes it is rather inefficient. A unit of this kind is shown in Fig. 97. It should always be installed in duplicate.

The capacity of a boiler-feed pump should be not less than twice the maximum rate of steam output to allow for sudden peaks, low water, etc. For reciprocating pumps the volumetric displacement should be three times the maximum rate of steam output, based on a piston speed in feet per minute of not more than ten times the square root of the number of inches in the length of the stroke.

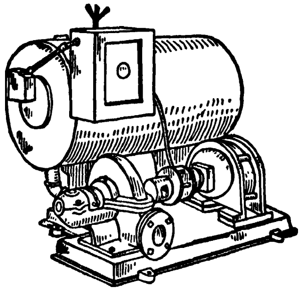


FIG. 97.—Motor-driven, boiler-feed pump.

142. Summary.—Of the several types of heating systems described in this chapter some are almost obsolete, but the student should be familiar with the general layout because they are often encountered in older buildings. In modern practice, steam heating systems have been narrowed down to not more than four types, which are used as follows:

Residences and *small* stores or *small* industrial buildings	Single-pipe system (Figs. 83 and 84) or Vapor system (Fig. 91)
Large industrial buildings	Overhead single-pipe system (Fig. 87) or Vacuum system
Large office buildings and all other large, well-finished buildings	Vacuum system

Bibliography

A.S.H. & V.E. "Guide," 1945, Chap. 14.
"Mechanical Equipment of Buildings," by Harding and Willard.
"Heating, Ventilating and Air Conditioning Fundamentals," by W. H. Severns.

CHAPTER VIII

PIPE, FITTINGS, VALVES, AND ACCESSORIES

143. Pipe.—The pipe used for the conveying of steam and water is made of cast iron, wrought iron, steel, copper, or brass. Because of the low tensile strength of cast iron, pipe made of this material is suitable only for low pressures and must have a relatively thick wall, but owing to its ability to withstand corrosion it is especially adaptable for use where it must be buried in soil. Cast-iron pipe is seldom used in heating work except occasionally for condensation-return piping. Certain alloyed cast irons, said to have greater corrosion resistance, have been introduced.

The pipe ordinarily used in heating and power plants is made from wrought iron or mild steel. Steel pipe is much more widely used than wrought-iron pipe at the present time because it is considerably lower in price and for many purposes equally desirable, although wrought-iron pipe is believed by some to be more resistant to corrosion.

In the manufacture of pipe the strips of metal, cut to the proper width, are drawn through a bell to the cylindrical form and the wedges are wedged together. In pipe of the smaller diameters a butt weld is used, and in the larger sizes a lap weld.

The dimensions of wrought-iron and steel pipe are given in Table 44. There are several schedules of wall thicknesses, and it should be noted that the outside diameters are uniform. The varying wall thicknesses are obtained by varying the inside diameters.

Schedule 40 corresponds for the most part to what has long been called "standard-weight pipe," and Schedule 80 corresponds to "extra-heavy" pipe. Those terms, however, are being eliminated. Schedule 40 pipe is the lightest weight that can be safely threaded. The lighter schedules are intended for welding or other types of joints that do not reduce the wall thickness. Schedule 40 pipe is the weight generally used in heating work.

TABLE 44.—DIMENSIONS OF WELDED AND SEAMLESS STEEL PIPE (American tentative Standard B36.10-1939, American Standards Association)

Nominal pipe size, in.	Outside diam., in.	Nominal wall thicknesses for schedule numbers, in.					
		Sched. 10	Sched. 20	Sched. 30	Sched. 40	Sched. 60	Sched. 80
1/8	0.405	. . . .	. . . . .	. . . . .	**0.068**	. . . . . .	**0.095**
1/4	0.540	. . . . .	. . . . .	. . . . .	**0.088**	. . . . . .	**0.119**
3/8	0.675	. . . . .	. . . . .	. . . . .	**0.091**	. . . . . .	**0.126**
1/2	0.840	. . . . .	. . . . .	. . . . .	**0.109**	. . . . . .	**0.147**
3/4	1.050	. . . . .	. . . . .	. . . . .	**0.113**	. . . . . .	**0.154**
1	1.315	. . . . .	. . . . .	. . . . .	**0.133**	. . . . . .	**0.179**
1 1/4	1.660	. . . . .	. . . . .	. . . . .	**0.140**	. . . . . .	**0.191**
1 1/2	1.900	. . . . .	. . . . .	. . . . .	**0.145**	. . . . . .	**0.200**
2	2.375	. . . . .	. . . . .	. . . . .	**0.154**	. . . . . .	**0.218**
2 1/2	2.875	. . . . .	. . . . .	. . . . .	**0.203**	. . . . . .	**0.276**
3	3.5	. . . . .	. . . . .	. . . . .	**0.216**	. . . . . .	**0.300**
3 1/2	4.0	. . . . .	. . . . .	. . . . .	**0.226**	. . . . . .	**0.318**
4	4.5	. . . . .	. . . . .	. . . . .	**0.237**	. . . . . .	**0.337**
5	5.563	. . . . .	. . . . .	. . . . .	**0.258**	. . . . . .	**0.375**
6	6.625	. . . . .	. . . . .	. . . . .	**0.280**	. . . . . .	**0.432**
8	8 625	. . . . .	0.250	**0.277**	**0.322**	0.406	**0.500**
10	10 75	. . . . .	0.250	**0.307**	**0.365**	**0.500**	0.593
12	12.75	. . . . .	0.250	**0.330**	0.406*	0.562*	0.687
14 O.D.	14.0	0.250	0.312	0.375	0.437	0.593	0.750
16 O.D.	16.0	0.250	0.312	0.375	0.500	0.656	0.843
18 O.D.	18.0	0.250	0.312	0.437	0.562	0.718	0.937
20 O.D.	20.0	0.250	0.375	0.500	0.593	0.812	1.031
24 O.D.	24.0	0.250	0.375	0.562	0.687	0.937	1.218
30 O.D.	30.0	0.312	0.500	0.625			

NOTE.—Thicknesses shown in boldface type in Schedules 30 and 40 are identical with thicknesses for "standard-weight" pipe in former lists; those in Schedules 60 and 80 are identical with thicknesses for "extra-strong" pipe in former lists.

Schedules 100 and above, for higher pressures, are omitted from this tabulation but are part of the complete standard.

* Twelve-inch pipe is not yet commercially available in Schedule 40 and Schedule 60 dimensions. The old "standard-weight" and extra-strong" standards with wall thicknesses of 0.375 and 0.500, respectively, are still in use.

TABLE 45.—PROPERTIES OF SCHEDULE 40 PIPE
(Conforming to A.S.A. Standard B36.10 for wrought-iron and wrought-steel pipe)

Nominal pipe size, in.	Outside diam. D, in.	Wall thickness t, in.	Inside diam. d, in.	Inside diam. squared d^2	Inside diam. fifth power d^5	Area of metal, sq. in.	Internal cross-sectional area		External surface per ft., sq. ft.	Weight of pipe per ft., lb.	Weight of water per ft. of pipe, lb.
							Sq. in.	Sq. ft.*			
⅛	0.405	0.068	0.269	0.0724	0.00141	0.072	0.0569	0.00040	0.106	0.25	0.0250
¼	0.540	0.088	0.364	0.1325	0.00639	0.125	0.1041	0.00072	0.141	0.43	0.0449
⅜	0.675	0.091	0.493	0.2430	0.02912	0.167	0.1909	0.00133	0.177	0.57	0.0830
½	0.840	0.109	0.622	0.386	0.09310	0.250	0.3039	0.00211	0.220	0.86	0.1317
¾	1.050	0.113	0.824	0.679	0.3799	0.333	0.5333	0.00371	0.275	1.14	0.2315
1	1.315	0.133	1.049	1.100	1.270	0.494	0.8639	0.00600	0.344	1.68	0.3744
1¼	1.660	0.140	1.380	1.904	5.005	0.669	1.495	0.01040	0.435	2.28	0.6490
1½	1.900	0.145	1.610	2.592	10.82	0.799	2.036	0.01414	0.497	2.72	0.8823
2	2.375	0.154	2.067	4.272	37.72	1.075	3.356	0.02330	0.622	3.66	1.454
2½	2.875	0.203	2.469	6.096	91.75	1.704	4.788	0.03322	0.753	5.80	2.073
3	3.5	0.216	3.068	9.413	271.8	2.288	7.393	0.05130	0.916	7.58	3.201
3½	4.0	0.226	3.548	12.59	562.2	2.680	9.888	0.06870	1.047	9.11	4.287
4	4.5	0.237	4.026	16.21	1,058	3.173	12.73	0.08840	1.178	10.8	5.516
5	5.563	0.258	5.047	25.47	3,275	4.304	20.01	0.1390	1.456	14.7	8.674
6	6.625	0.280	6.065	36.78	8,206	5.584	28.99	0.2006	1.734	19.0	12.52
8	8.625	0.322	7.981	63.70	32,380	8.396	50.03	0.3474	2.258	28.6	21.68
10	10.75	0.365	10.02	100.4	101,000	11.90	78.85	0.5475	2.814	40.5	34.16
12†	12.75	0.406	11.938	142.5	242,470	15.77	111.93	0.7773	3.338	53.6	48.50
14 O.D.	14.0	0.437	13.126	172.3	389,638	18.61	135.32	0.9397	3.665	63.8	58.64
16 O.D.	16.0	0.500	15.000	225.0	759,375	24.35	176.72	1.2272	4.189	82.8	76.58
18 O.D.	18.0	0.562	16.876	284.8	1,368,820	30.79	223.68	1.5533	4.712	105	96.93
20 O.D.	20.0	0.593	18.814	354.0	2,357,244	36.15	278.00	1.9305	5.236	123	120.46
24 O.D.	24.0	0.687	22.626	511.9	5,929,784	50.31	402.07	2.7921	6.283	171	174.23

Underscored wall thicknesses are identical with thicknesses for standard-weight pipe in former tables.

* This column also represents the contents in cubic feet per foot of length.

† Twelve-inch pipe is not yet commercially available in Schedule 40 dimensions. The old "standard-weight" pipe with wall thickness 0.375 in. is still in use.

Table 45 gives the dimensional properties of Schedule 40 pipe for convenience in calculations.

144. Pipe Threads.—In order that they may be screwed to a tight joint, pipe threads are made with a taper of 1 in 32 with the axis of the pipe, and the threads in the fittings are tapped to the same taper. Pipe threads are made to conform to the American standard, illustrated in Fig. 98, which calls for a thread having a 60-deg. angle, with the top and bottom slightly flattened. The number of threads per inch varies for the different sizes of pipe.

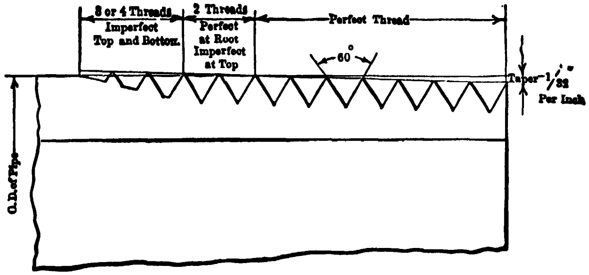


FIG. 98.—American standard pipe thread.

145. Fittings.—The common forms of screwed fittings used in heating work are shown in Fig. 99. All except the nipples and ordinary couplings are made of cast iron. In designating reducing tees, the size of the openings opposite each other is given first, and then the size of the branch opening. For example, the reducing tee in Fig. 99 is a 1¼- by 1- by ½-in. tee. A special elbow, called a pitched elbow, is designed for and sometimes used in heating work. It has an angle of slightly more than 90 deg. and is used to connect a vertical pipe with a basement "runout," which is sloped for proper drainage. Pitched elbows carry an identifying mark. They are used only in the best class of work or when definitely specified.

For pressures over 125 lb., an extra-heavy pattern is available, suitable for working pressures up to 250 lb. Extra-heavy fittings are made with a greater wall thickness and are of larger dimensions throughout.

Where screwed fittings are used, provision should be made, at intervals in the line, for disconnecting the piping for repairs, etc. Right-and-left couplings, or unions, are used for this purpose. The former, as the name indicates, are couplings tapped at one end with a left-hand thread, so that both threads can be screwed up simultaneously. Longitudinal ridges are

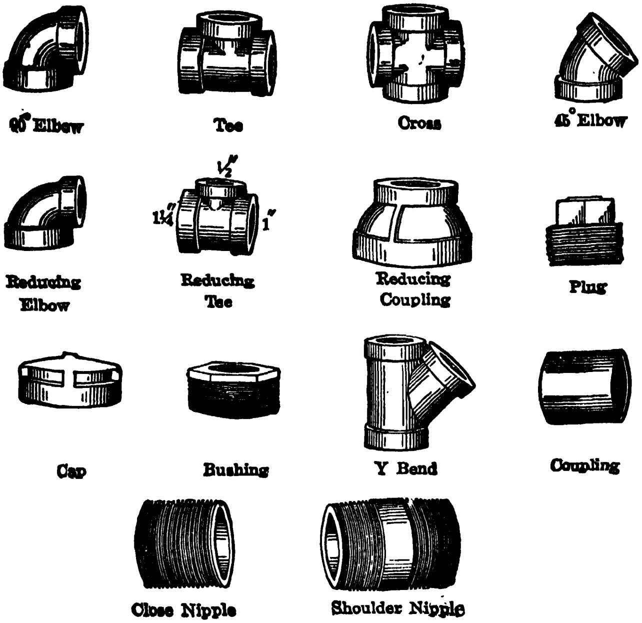


FIG. 99.—Screwed fittings.

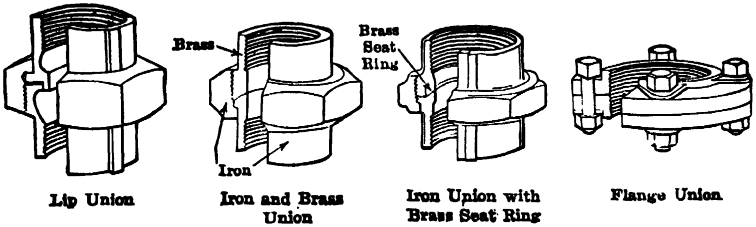
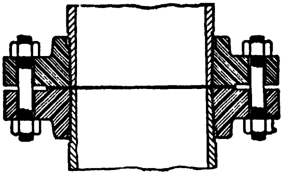


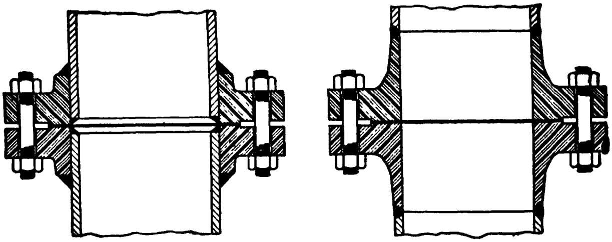
FIG. 100.—Pipe unions.

cast on right-and-left couplings so that they can be identified after installation.

For pipe sizes up to 2 in., nut unions, consisting of two pieces screwed to the ends of the pipe and held together by means of a threaded nut, are used. Flange unions are used with larger sizes of pipe. In Fig. 100 are shown these various types of pipe connections. The ground-joint union is superior to the gasket union in that it can be disconnected repeatedly without trouble, whereas the gasket in the latter type must be frequently replaced.



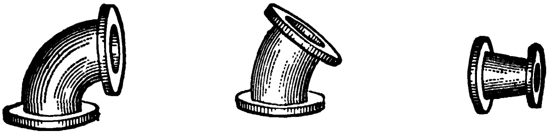
Screwed Flanges



Welded-on Flanges

Welding Neck Flanges

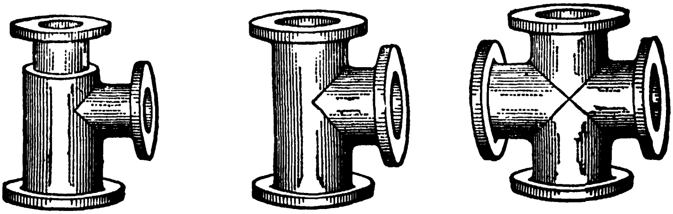
FIG. 101.—Pipe flanges.



90° Elbow

45° Elbow

Reducer



Reducing Tee

Tee

Cross

FIG. 102.—Flanged fittings.

In heating work, piping of the larger sizes (over 3 or 4 in.) is usually designed with flanged connections, in order that any section of pipe or any fitting may be readily removed. With screwed fittings it is necessary, in order to remove any member, to take down all of the line from the nearest union or flanged connection. Flanges are commonly screwed to the pipe or welded, as shown in Fig. 101.

Some forms of standard-weight flanged fittings are shown in Fig. 102. These fittings are suitable for pressures up to 125 lb. There is an extra-heavy pattern of flanges and flanged fittings, which differs both in general dimensions and in the number and spacing of bolts.

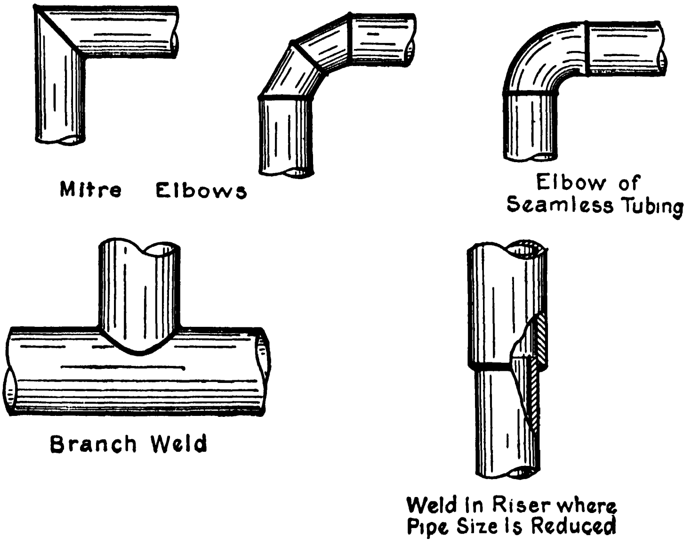


Fig. 103.—Welded construction.

In bolting flanged fittings together it is necessary to insert a gasket between the faces in order to ensure a tight joint. Gaskets are made of sheet rubber for water- and low-pressure steam lines; for high-pressure lines, gaskets of corrugated copper or of various compositions containing asbestos are used. Gaskets are preferably cut in a plain ring to fit inside the flange bolts.

146. The Welded Joint.—Fusion welding has come to be very frequently used in heating work. As a method of assembling piping systems it has many advantages. In the larger sizes it is considerably cheaper than the use of fittings. It not only saves the cost of fittings, bolts, and gaskets but reduces the labor cost somewhat; the finished piping is better looking, and insulation can be more easily and cheapy applied. Welding is not economical with small sizes of pipe—below, say, 3 in.—because of the difficulty of welding thin metal and because of the small cost of screwed fittings.

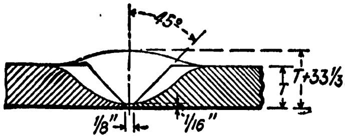


Fig. 104.—Method of beveling for oxyacetylene weld.

Several forms of welded construction are shown in Fig. 103. The steel elbow of seamless tubing, which is shown, is often used to advantage. The edges to be welded are beveled at an angle of 45 deg., as shown in Fig. 104.

Welding for heating work is done with either the oxyacetylene torch or the electric arc, each method having its advocates.

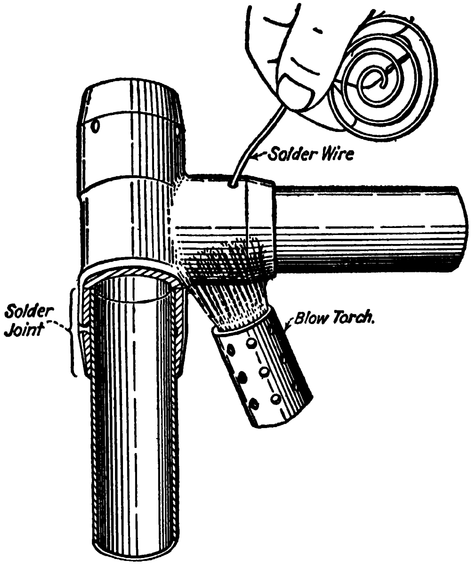


Fig. 105.- Copper tubing with sweated joint.

147. Brass and Copper Pipe.—Brass pipe is used occasionally in condensation-return lines where a material more resistant to corrosion than wrought iron or steel is required. Brass pipe is manufactured in sizes and wall thickness corresponding to wrought-iron and steel pipe, and cast-brass fittings are available.

Copper tubing is coming into use to some extent in heating work. It is used with special valves and fittings, into which it is made tight by a sweated joint, one form of which is shown in Fig. 105. Since the tubing has no threads cut into it, a much thinner wall is used, and this saving in weight largely compensates for the higher cost per pound of copper as compared with steel. In small sizes the difference in cost is not so great as might be expected.

The advantages of copper tubing for heating work are its lesser susceptibility to corrosion and its neatness in appearance.

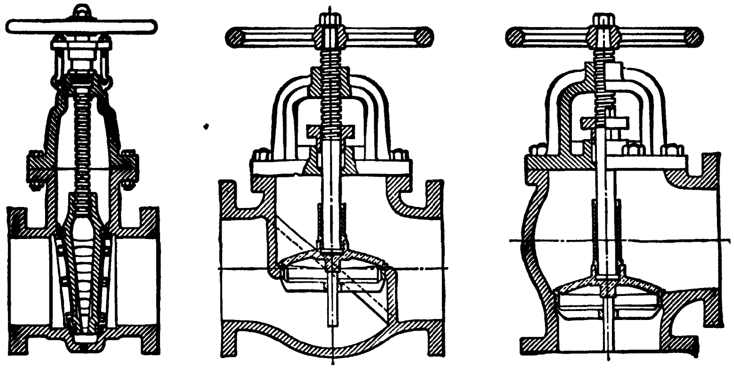
Table 46 gives the dimensions of tubing used in heating work.

TABLE 46.—DIMENSIONS OF COPPER TUBING

Type M 200-lb. nonshock working pressure or low-pressure steam				Type L 250-lb. nonshock working pressure or 65-lb. steam pressure and not over 310 deg. (with soldered fittings)			
Nominal pipe size	Outside diam.	Decimal equiv. of wall thickness	Actual inside diam.	Nominal pipe size	Outside diam.	Decimal equiv. of wall thickness	Acutal inside diam.
¼	⅜	0.025	0.325	⅜	½	0.035	0.430
⅜	½	0.025	0.450	½	⅝	0.040	0.545
½	⅝	0.028	0.569	⅝	¾	0.042	0.666
¾	⅞	0.032	0.811	¾	⅞	0.045	0.785
1	1⅛	0.035	1.055	1	1⅛	0.050	1.025
1¼	1⅜	0.042	1.291	1¼	1⅜	0.055	1.265
1½	1⅝	0.049	1.527	1½	1⅝	0.060	1.505
2	2⅛	0.058	2.009	2	2⅛	0.070	1.985
2½	2⅝	0.065	2.495	2½	2⅝	0.080	2.465
3	3⅛	0.072	2.981	3	3⅛	0.090	2.945
3½	3⅝	0.083	3.459	3½	3⅝	0.100	3.425
4	4⅛	0.095	3.935	4	4⅛	0.110	3.905
5	5⅛	0.109	4.907	5	5⅛	0.125	4.875
6	6⅛	0.122	5.881	6	6⅛	0.140	5.845
8	8⅛	0.170	7.785	8	8⅛	0.200	7.725

148. Valves.—In Fig. 106 are shown the various types of iron-body valves. The gate valve is the form ordinarily used in steam piping. Globe valves are not permissible in horizontal steam lines, as they are so constructed as to dam up the water and cause it to accumulate in the bottom of the pipe; but on vertical steam pipes and on water pipes they are permissible and are especially desirable when the flow of steam or water is to be throttled. The angle valve is a very good type for locations where it can be used.

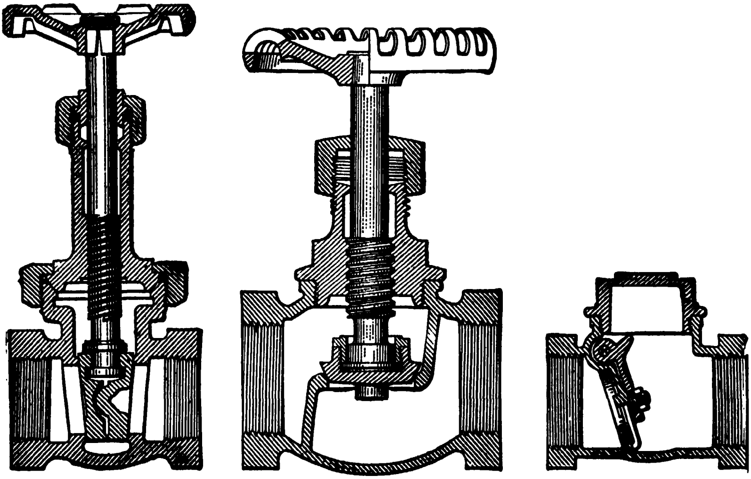
Valves in sizes up to 3 in. are usually made entirely of brass. The larger sizes are usually made of cast iron, with the gates and seats faced with bronze to give a noncorroding surface. The bronze mountings can be replaced when worn. The cover, or bonnet, of these larger valves is bolted instead of screwed to the


Iron-body gate valve, nonrising stem.

Iron-body globe valve, rising stem.

Angle valve.

FIG. 106.—Iron-body valves.


All-brass gate valve.

All-brass globe valve.

Swing check valve.

FIG. 107.—Brass valves.

body. Gate valves are made with either a rising or nonrising stem. With a rising-stem valve the amount to which the valve is opened is always apparent, and this is often of great advantage, but the space occupied by the valve is somewhat greater. Figure 107 illustrates brass valves.

Check valves are frequently used in heating work. The swing check, illustrated in Fig. 107, is the most satisfactory form.

The ordinary radiator valve for steam is of the angle pattern and is provided with a union for connecting to the radiator, as shown in Fig. 108. The valve disk is made of hard rubber and is renewable. These valves are also made in the "corner" pattern, having a side inlet.

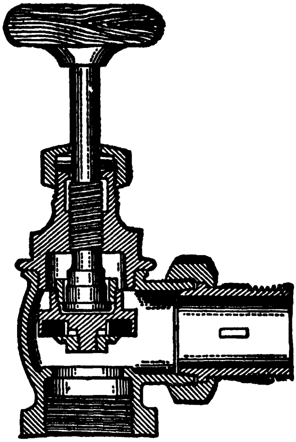


FIG. 108.—Ordinary radiator valve.

The stem of the ordinary radiator valve is packed with a soft stranded packing to prevent leakage. The packing is seldom permanently tight, however, and the resulting leakage is often a source of considerable annoyance. In the more modern valves the packing is replaced by a grooved hard-rubber washer, which is held against a seat by a spring. The construction of these so-called "packless" valves was discussed in the previous chapter, Art. 134.

149. Pipe Covering.—The piping of a heating system that is not intended to serve as a radiating surface is insulated with some material of low heat conductivity. Most insulating materials owe their useful property to air, enclosed in extremely small volumes. If the material is to be an efficient insulator, these air volumes must be so minute that the circulation of the air in them is reduced to a minimum, and in addition the material itself must be of low conductivity. A satisfactory pipe covering must also be able to withstand the effect of high temperature and vibration and to retain its insulating qualities throughout a long period of years. Pipe coverings are made of magnesia, asbestos, infusorial earth, glass wool, mineral wool, hair felt, wool felt, and other materials. These substances form the basic element and are usually combined with other materials for mechanical reasons.

Probably the most widely used insulating material is magnesium carbonate. It is in the form of a white powder, and some fibrous material such as asbestos fibers must be used with it as a binder, the aggregate being molded into blocks or into segments curved to fit the various sizes of pipe. Infusorial earth, which consists of the siliceous shells of minute organisms, is also combined with various binding materials to form a very efficient covering.

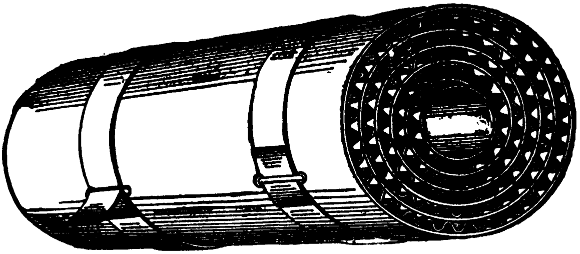


FIG. 109.—Cellular pipe covering.

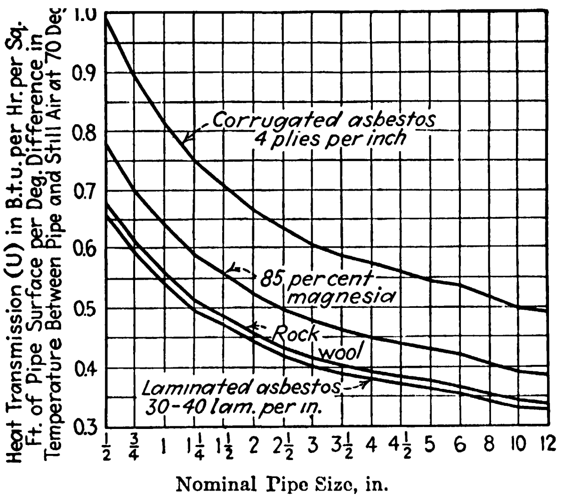


FIG. 110.—Heat loss through various kinds of pipe insulation, 1-in. thickness. Steam at 227.1° (5-lb. pressure).

Many forms of pipe covering are made of asbestos in combination with some cellular material. The compound is rolled into sheets, and the covering is built up in corrugations so as to enclose air spaces. Although not very efficient or durable, this type of covering is widely used because of its low price. Figure 109 shows this type. The heat loss through various kinds of insulation is shown in Fig. 110 for 1-in.-thick insulation. The heat loss from bare pipe is given in Table 47.

TABLE 47.—HEAT LOSS FROM HORIZONTAL BARE IRON PIPE[1]
[Steam temperature 227.1° (5 lb. pressure); still air at 70°]

Nominal Pipe Size, In.	Heat Loss, B.t.u. per Lin. Ft. per Deg. Diff. between Pipe and Air per Hour
½	0.656
¾	0.781
1	0.953
1¼	1.184
1½	1.335
2	1.637
2½	1.937
3	2.301
3½	2.585
4	2.873
5	3.493
6	4.115
8	5.270
10	6.551
12	7.670

[1] Mellon Institute. This table is reproduced exactly, but it is not to be assumed that the values given would be found to exist in the field with the exactness implied by the four digits.

It is nearly always desirable to provide insulation on the boiler and on the basement and attic mains in a heating system. It is usually desirable also to cover the supply risers, because they would otherwise give off heat continuously whether it were needed or not. Return pipes are seldom covered in a system equipped with thermostatic traps.

150. Air Valves.—In the ordinary steam heating system the air that fills the radiators when they are cold is forced out by the entering steam through some form of air valve installed on the end of the radiator opposite the supply connection. These air valves may be simply hand-operated cocks, which must be opened whenever the radiator is turned on, but the many forms of air valves that allow the air to escape but close automatically when steam reaches them are greatly to be preferred. Automatic air valves are also designed to close when flooded with water, as they sometimes are when a radiator does not drain properly or becomes filled with water because of a leaky inlet valve. A common design is shown in Fig. 111.

151. Traps.—A steam trap is a device whose function is to drain the water from a steam pipe, separator, or radiator,

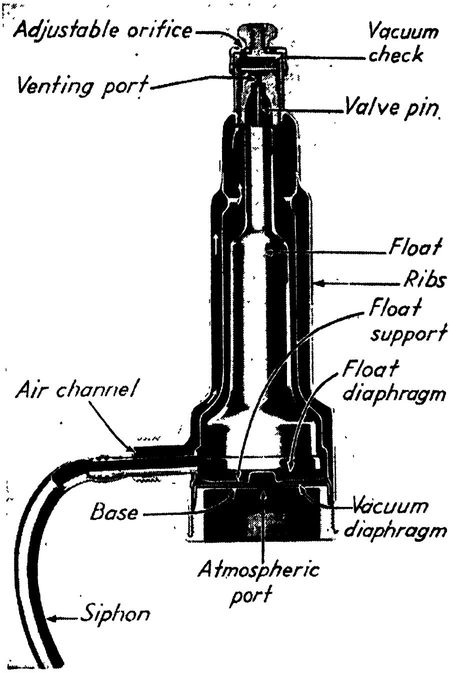


FIG. 111.—Radiator air valve. (*Hoffman Specialty Company.*)

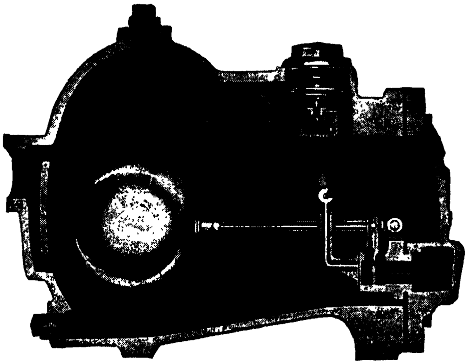


FIG. 112.—Float trap. (*James P. Marsh Corp.*)

without allowing steam to escape. For radiators, special traps of the float, or thermostatic, form described in Chap. VII are used. For draining steam lines and separators, there are two kinds of traps in use, designated as "float" and "bucket" traps. The float trap consists of a receiver having a discharge valve controlled by a float in such a way that a rise of the water level from an influx of water causes the float to open the valve, allowing water to be discharged by the pressure of the steam until the water level is lowered to its normal point. One design of float trap is shown in Fig. 112. It contains an automatic air-vent valve operated by a thermostatic element.

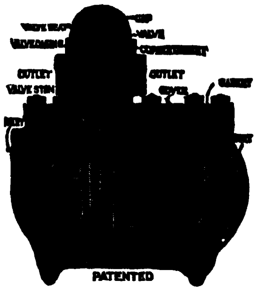


FIG. 113.—Bucket trap. (*C. E. Squires Company.*)

The bucket trap (Fig. 113) consists of a chamber containing a bucket which is floated by the water in the chamber. To the bucket are attached the valve stem and valve. The water flowing into the trap enters and fills the bucket, finally causing it to sink and thereby opening the discharge valve. The steam pressure forces the water out through the valve and empties the bucket, which rises and closes the valve.

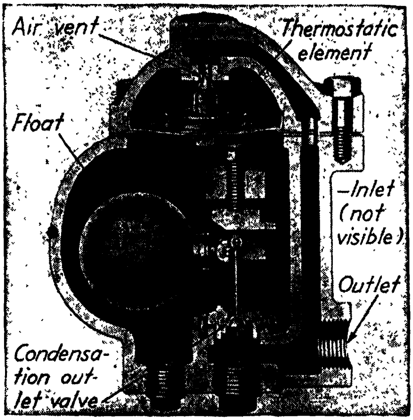
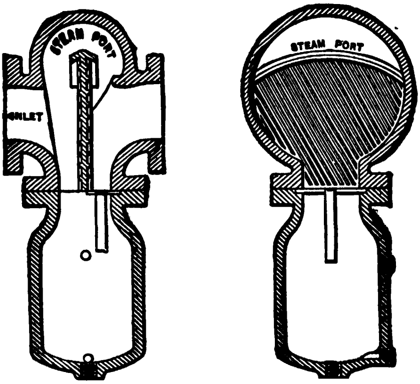


FIG. 114.—Blast trap. (*Warren Webster and Company.*)

It is possible to lift the condensation by means of a trap to a height approaching that equivalent to the steam pressure, that is, about 2.3 ft. per pound pressure. It is better, however, if possible, to locate the trap so that it will discharge by gravity.

For draining the steam coils of a fan system a so-called "blast trap" such as that shown in Fig. 114 is generally used. The amount of condensation in a fan heater is often very large—more than can be handled by an ordinary thermostatic radiator trap. A blast trap of this type has a relatively large valve operated by a float and a thermostatic valve that serves to vent the air. This combination provides ample venting for both air and condensation.



SIDE SECTION END SECTION

Fig. 115.—Steam separator.

152. Separators.—The function of a steam separator is to remove condensation from steam lines. The separator accomplishes this by abruptly changing the direction of flow of the steam and thereby causing the entrained water to be thrown out by its momentum against a suitably designed baffle, usually having a series of grooves that conduct the water into a receiver below. The water is discharged through a trap or seal. This form of separator is illustrated in Fig. 115. Separators are placed in the exhaust line from pumps and reciprocating engines, where they remove the oil as well as the water from the steam. In choosing a separator, care must be taken to select a size corresponding to the quantity of steam flowing rather than to the size of the pipe line, for a certain velocity through the separator is necessary to ensure the elimination of the water.

153. Reducing Valves.—Occasionally steam is supplied to a building at a pressure much higher than is necessary or desirable for heating purposes, making it necessary to employ a reducing valve, a simple form of which is illustrated in Fig. 116. The pressure on the reduced-pressure side of the valve is transmitted through the balance pipe to the under side of the diaphragm, tending to close the valve. The force thus exerted is balanced by that due to the weights *W*, *W*, and the valve will assume such a position that just enough steam will pass through it to maintain the required pressure on the reduced-pressure side, a pressure governed by the position of the weights on the lever arm and changeable as desired by shifting these weights. The valve shown in Fig. 116 is double seated, so that its movement is nearly independent of the steam pressure on either side of the disks, and is controlled almost solely by the reduced pressure acting on the diaphragm. Reducing valves should be installed with a by-pass so that they can be removed for repairs without interruption of the steam supply.

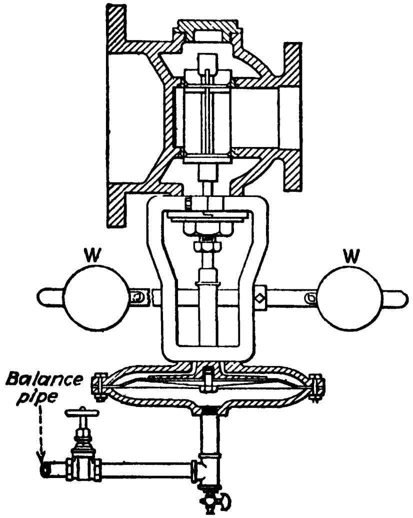


Fig. 116.—Reducing valve, direct-acting type.

Figure 117 shows a reducing valve of the pilot-operated type. Where the reduced pressure is above about 30 lb., the rubber-diaphragm type is not suitable. The valve of Fig. 117 has a metal diaphragm that operates a small pilot valve admitting steam above the piston, which operates the main valve. There are numerous variations of this design, but it is generally typical of valves for higher pressure service or where accurate control of the reduced pressure is necessary.

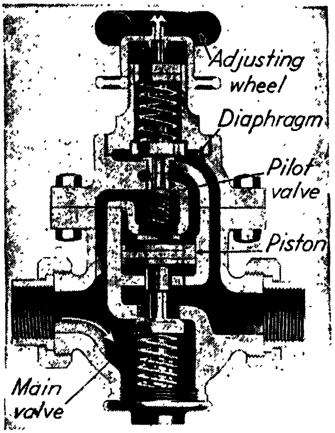


FIG. 117.—Pilot-operated reducing valve. (*Illinois Engineering Company.*)

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Various articles on the welding of piping, *American Welding Society Journal*

Problem

A low-pressure steam main in a heating system consists of 75 ft. of 8-in. pipe, 91 ft. of 6-in., 63 ft. of 4-in., and 37 ft. of 2-in. It is in service 11 hr. per day for 230 days in the year. Steam costs 30 cents per 1,000 lb. What would be the gross saving by insulating the pipe with (*a*) corrugated asbestos insulation and (*b*) laminated asbestos insulation?

CHAPTER IX

STEAM PIPING

154. Principles Involved in Piping Design.—The elementary forms of the different systems of steam heating were shown diagrammatically in Chap. VII. The actual arrangement of the piping will be studied in this chapter. The success of a heating system depends very largely upon the proper size and correct arrangement of the pipes and fittings, and it is necessary to give careful attention to many details that might at first thought be considered insignificant. One reason why heating-system piping requires particularly careful design is that it must convey three different fluids—steam, water, and air—which have different properties and one of which is often flowing in a direction opposite to the others in the same pipe.

In designing and installing a steam heating system, attention must be given to the following fundamental requirements: (1) provision for thermal expansion, (2) provision for flow of condensation, (3) provision for removal of air, and (4) selection of proper pipe sizes.

155. Expansion.—Perhaps the most important consideration is the proper provision for the linear expansion of the pipes. When steam is turned into or shut off from a system of piping, a change of temperature of the pipe amounting to 140 to 170° takes place, and provision must be made for allowing the resulting change of length to occur without putting excessive strains on the fittings. The curve in Fig. 118 shows the theoretical expansion of pipe due to an increase in temperature from 60° to the temperature corresponding to various steam pressures. The temperature of 60° is assumed to be that at which the piping is originally installed. For low-pressure piping a rough rule is to allow 1½ *in. of expansion per* 100-*ft. length of pipe.*

There are, in general, three ways in which the expansion in a system of piping may be absorbed: (*a*) by the turning of some of the threaded joints, (*b*) by the bending of the pipes, and (*c*) by the use of special devices designed to absorb the movement.

The absorbing of the expansive movement by the turning of threaded joints is permissible only in low-pressure piping work. Continued twisting of a threaded joint will in time often result in a leak, particularly when the pressure is high. In heating work it is common practice to depend upon this method of caring for expansion, although it is questionable whether the threaded joints actually turn as intended after corrosion has taken place. On the whole, the other methods, *b* and *c*, are more satisfactory. Examples of both of these will be described later.

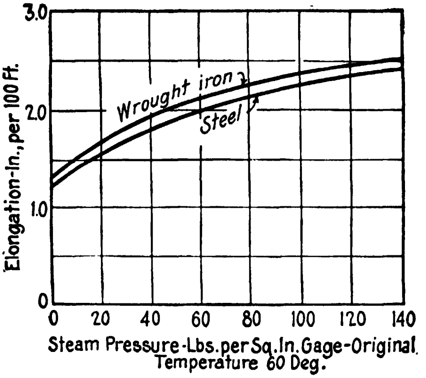

FIG. 118.—Elongation of pipe with various steam pressures.

156. Drainage.—Adequate provision for the flow of condensation is always important in pipes carrying saturated steam. In heating systems this is particularly true because very often a pipe is carrying not only the condensation that forms in the pipe itself but also condensation from radiators. Wherever possible, it is best to arrange the piping so that the water will travel in the same direction as the steam, but in single-pipe systems the radiator condensation must flow in the direction opposite to the steam flow, and unless the pipes are of ample size so as to give low steam velocities and are properly sloped, the water cannot flow back freely and will be held up in the radiators.

The pitch of pipes should be continuous, and any unavoidable pockets should be drained in some manner. A common error is the use of fittings that form water pockets like those caused by the reducing tee shown in Fig. 119. The reduction in pipe size should be made by using an eccentric reducer, as in Fig. 120.

Steam mains that pitch away from the boiler are provided with drip connections at the extreme end, and an air-vent valve is installed as in Fig. 121. The installation of air vents at the ends of mains and the tops of long upfeed risers is desirable in order to promote rapid circulation. Drip connections from different mains should not be tied together above the water line of the boiler but should be brought down well below the water line, so that the slightly different pressures will not cause a backflow through one of them.

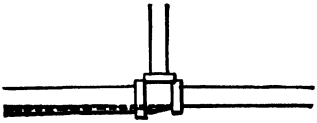

Fig. 119.—Wrong method.

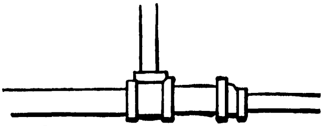

Fig. 120.—Correct method.

157. Mains and Branches.—Horizontal mains are usually anchored at the boiler and allowed to expand freely from that point. The amount of movement of any point along the length of the pipe is evidently proportional to its distance from the fixed point. In connecting risers and branches the movement of the mains is best taken care of by either of the arrangements in Figs. 122 and 123. As the main moves longitudinally the branch pipe bends, and in some cases the threaded joints *C*, *C* probably turn slightly. The arrangement of Fig. 123 is somewhat the better, as the 45-deg. elbow offers less resistance to the flow of steam than the 90-deg elbow in Fig. 122. The expan-

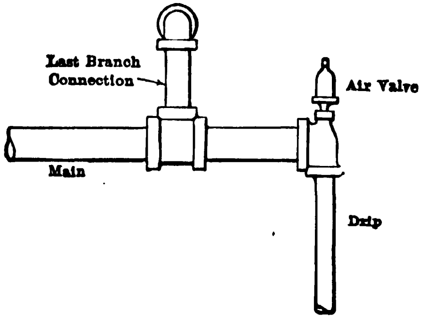

Fig. 121.—Method of dripping end of steam main.

sion of the horizontal branch is taken care of by the spring of the riser, an arrangement that is quite permissible, as such branches are rarely over 10 ft. long.

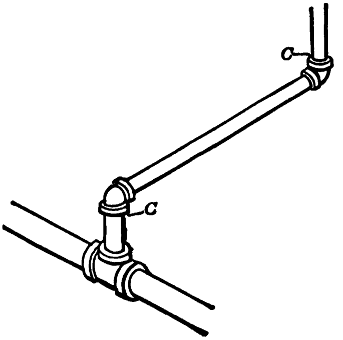

FIG. 122.

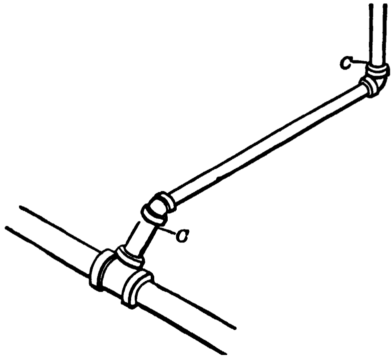

FIG. 123.

FIGS. 122 and 123.—Methods of connecting branches.

158. Risers.—In small buildings where the supply mains are in the basement, the expansion of the risers is usually downward, and the movement is taken care of by the spring of the branches (Figs. 122 and 123). In larger buildings where there is a main in the attic, the risers are anchored near the middle, and the expansion takes place in both directions; and in very tall buildings several anchor points and expansion elements are necessary. When the expansion is too great to be handled by an ordinary

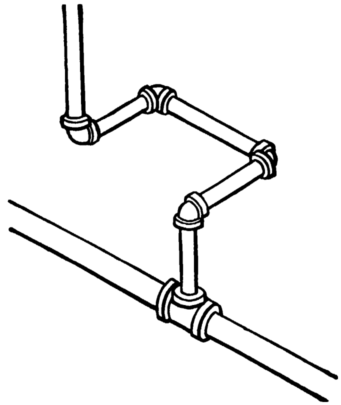

FIG. 124.—Flexible connection for riser.

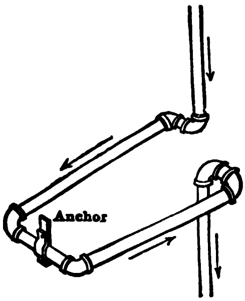

FIG. 125.—Expansion loop for riser.

branch connection, the arrangement in Fig. 124 may be used. This gives a true swivel joint and is especially serviceable when the basement main must be installed near the foot of the risers. Its disadvantage is the resistance to the steam flow offered by the fittings.

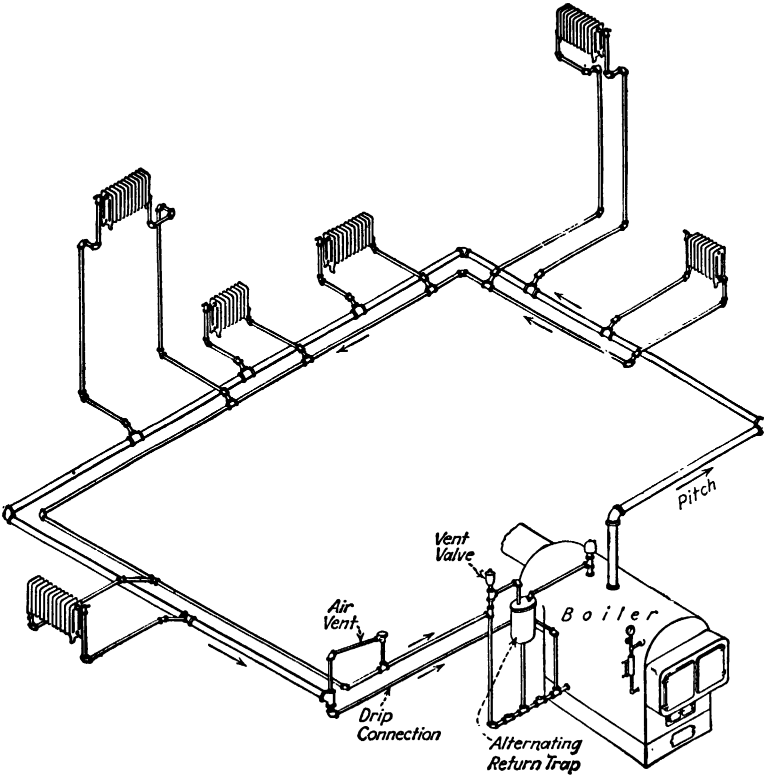

FIG. 126.—Vapor system for a small building.

The branch connection shown in Fig. 123 will easily take care of the expansion of risers about four stories high; and that in Fig. 124, about eight stories. For taller buildings one or more loops of the form shown in Fig. 125 are installed in the middle of each riser. Such an expansion loop is easily capable of handling a length of riser of at least four stories in either direction and gives good flexibility. Space is required in the furring to conceal the loop. Where space does not permit the use of any sort of expansion loop, some form of expansion joint especially designed for the purpose is used. The slip joint, which consists of a sleeve sliding in a packed gland, is one form.

159. Piping Systems.—The complete piping for some of the types of systems shown diagrammatically in Chap. VII will now be considered. Figure 126 shows a vapor system for a residence or other small buildings. The supply main extends around the basement at a proper distance from the outside walls to allow for the branches to the risers. The return main is placed alongside it and pitching in the same direction. In many cases it is desirable to run separate supply and return mains to different parts of the basement, and the best layout is usually that which uses the least length of pipe.

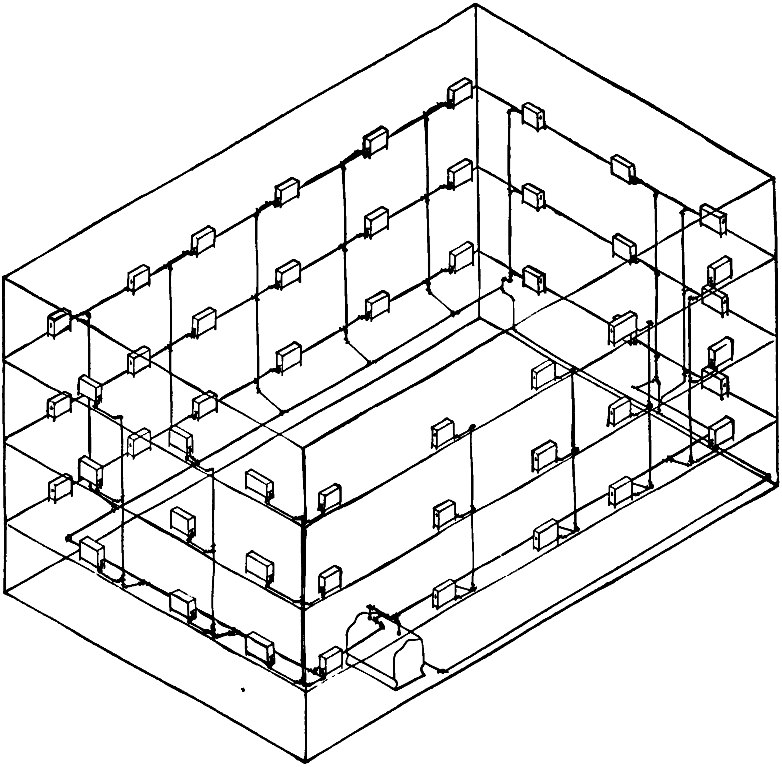

FIG. 127.—Single-pipe upfeed system.

Figure 127 shows the complete piping for a single-pipe upfeed system in a three-story building. This particular system has two supply mains, pitching away from the boiler and dripped at their ends into the return main.

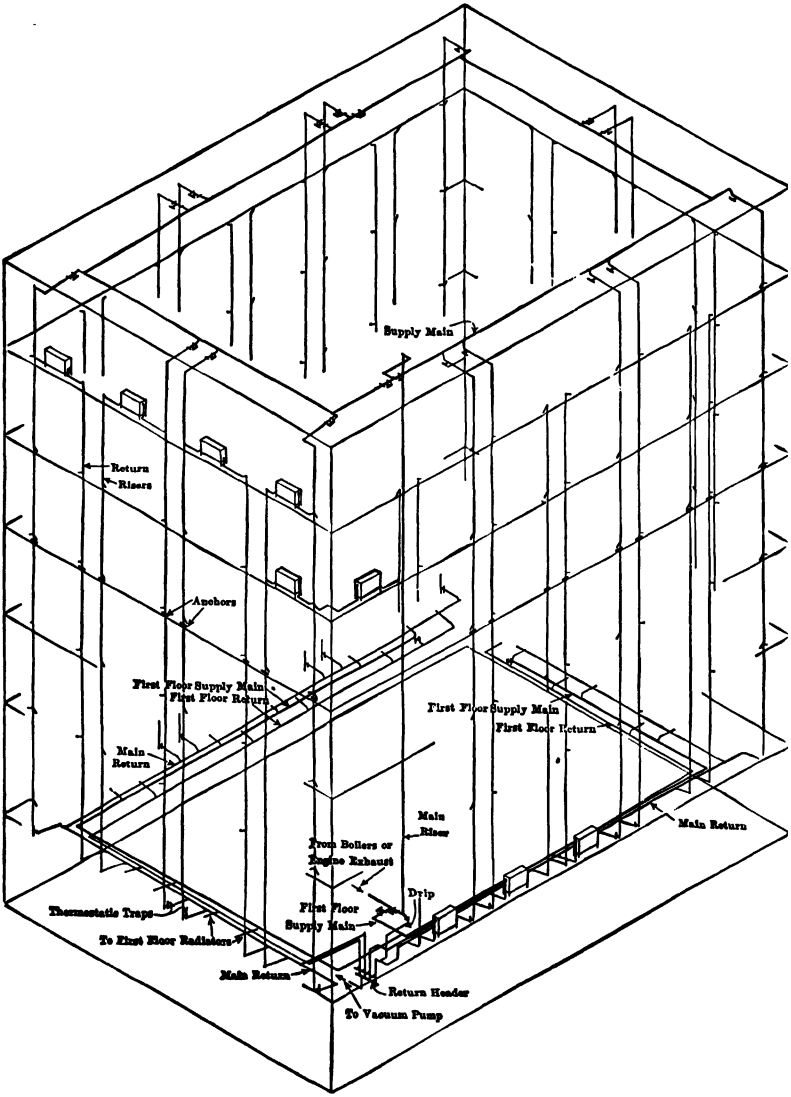

FIG. 128.—Overhead vapor or vacuum system.

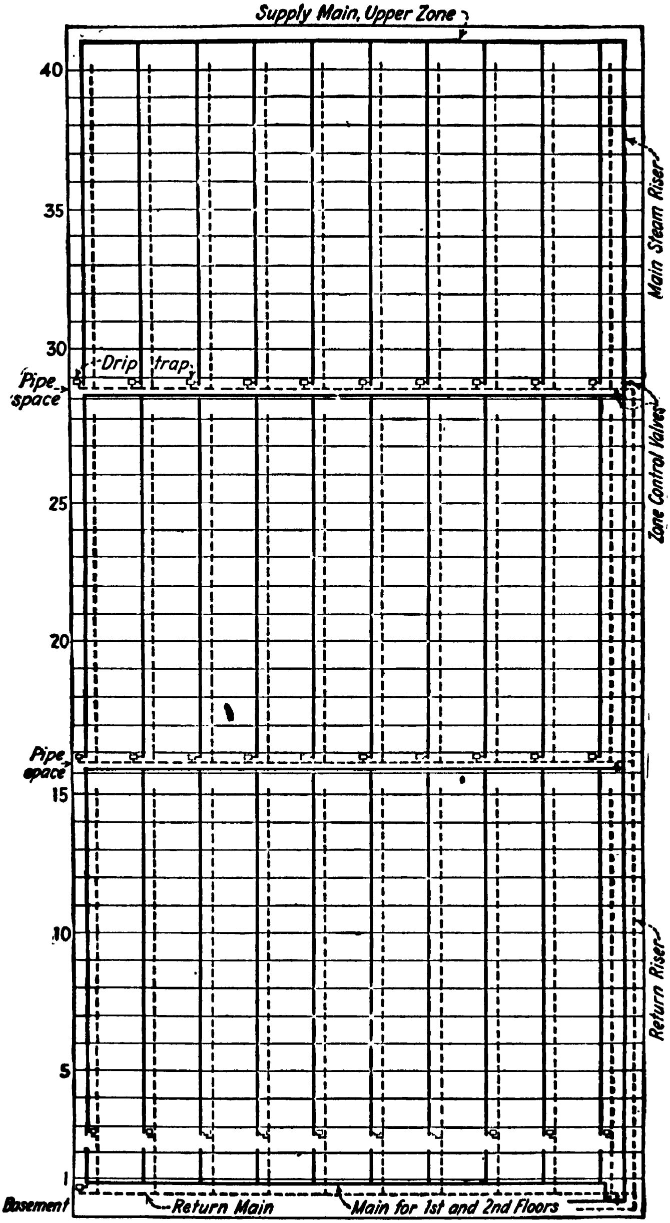

FIG. 129.—Piping in a tall building.

Figure 128 shows an overhead vapor or vacuum system in a tall building. Return risers extend from the top-floor radiators to the basement, where they tie into the main return line. In large buildings the first floor is often divided into small stores, which require heat at times when none is needed in the remainder of the building, and vice versa, making it desirable to install a separate main to supply the first-floor radiators and arranged so that it can be controlled independently of the main heating system. This scheme also has the advantage of making it much easier to install connections to the first-floor radiators, which are often so located that it is difficult to reach them from the risers. In tall buildings it is better to feed the risers from the bottom as well as from the top, and a supply main is installed in the basement for that purpose. In extremely tall buildings it is often the practice to divide the building into zones about ten stories high, each having its own supply main. Figure 129 shows a tall building zoned in this manner. The supply mains are carried in pipe spaces located between floors and of less height than the usual floor-to-ceiling dimensions. In a tall building pipe spaces of this sort are very convenient to take care of air ducts, wiring conduit, and other things in addition to the heating pipes. Zoning of a heating system according to the sun and wind exposures of its several sides is sometimes done to make possible an economical control of the heat supply. Separate control valves are provided for the different sections.

160. Return Piping.—Return pipes of any kind of steam system should be designed with ample provision for expansion, as they may at times be heated to steam temperatures. Dry-return mains should be given a pitch of at least 1 in. in 20 ft. toward the boiler. Wet-return mains should also be pitched toward the boiler so that they may be entirely drained of water when necessary. Return pipes should never be buried in the ground without protection. When it is necessary to conceal them, the best plan is to arrange them in trenches with removable cover plates. An alternate scheme is to cover them with cylindrical tile with cemented joints that will keep the water out. When buried in soil, return pipes corrode and deteriorate very rapidly unless built of special materials.

161. Vapor and Vacuum Systems.—In a vapor system, since the return lines are at atmospheric pressure, the water will build up in the return leg to a height, above the water in the boiler, equivalent to the pressure in the boiler. In order to prevent the return mains from becoming flooded, the distance from the water line of the boiler to the horizontal return main should be as great as possible, never less than 2½ ft. unless an alternating return trap or a boiler-feed pump is used. In some cases it is necessary to place the boiler in a pit below the basement floor. The supply main of a vapor system can often be dripped at the end into the return main through a thermostatic trap. This method, however, necessitates starting the return main at an elevation below the end of the supply main, which, with the necessary pitch toward the boiler, may bring it very close to the water line. A better arrangement is to install a separate drip line from the end of the supply main, which allows the return main to be placed much higher. This arrangement is shown in Fig. 130, the dotted line representing the necessary elevation of the return main if the drip line is omitted.

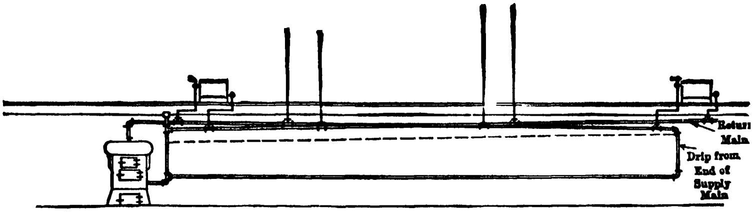

FIG. 130.—Method of dripping supply main when basement is shallow.

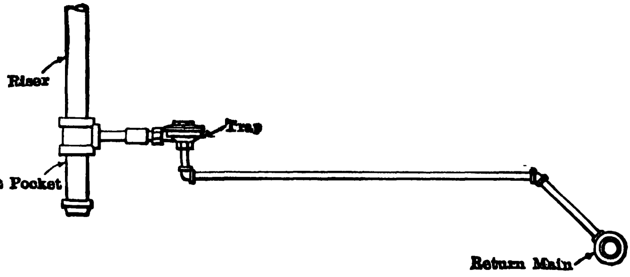

FIG. 131.—Drip connection to riser in a vapor or vacuum system.

In an overhead vapor or vacuum system each riser is dripped at the bottom through a thermostatic trap, as in Fig. 131. In order to catch the dirt and scale that would clog the trap, a dirt pocket should be provided, consisting of a short capped pipe. Steam mains are dripped into the return line in a similar manner. By-passes are sometimes provided for the most important traps to enable them to be easily cleaned or inspected, and dirt strainers are also sometimes used.

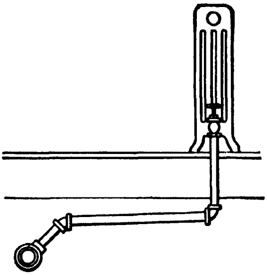

FIG. 132.—Connection to a first-floor radiator.

162. Radiator Connections.—The connection to a radiator must be sufficiently flexible for the main or riser to be perfectly free to expand without straining the fittings. They must also be designed to allow the radiator to drain properly and must be free from water pockets. Figure 132 shows the proper method of connecting a first-floor radiator to a basement main.

The connections to a radiator of a vapor or vacuum system are shown in Fig. 133. These connections are also very flexible, and the use of 45-deg. elbows reduces the frictional resistance.

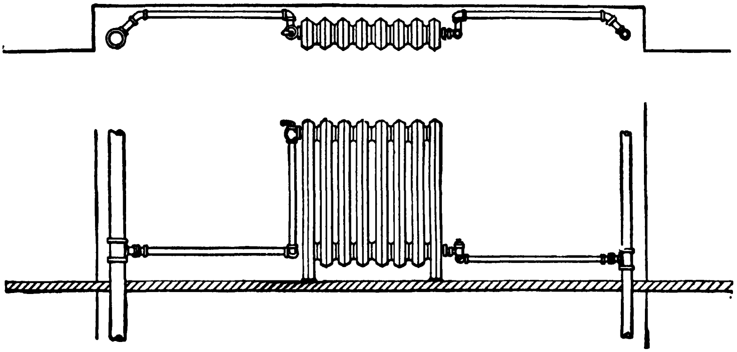

FIG. 133.—Radiator connections—vapor or vacuum system.

163. Pitch of Pipes.—The slope, or pitch, of pipes is obviously very important, and a greater pitch is required if condensation is to be drained against the flow of steam than otherwise. The *minimum* pitch for various parts of the piping should be as follows:

	Pitch Not Less Than
Mains, condensation flowing with steam	1 in. in 20 ft.
Mains, reverse flow of condensation	1 in. in 10 ft.
Branches to risers and radiators	1 in. in 10 ft.
Dry-return mains	1 in. in 20 ft.
Wet-return mains	1 in. in 20 ft.

A greater amount of pitch is not objectionable, except that in long runs it may bring the pipes too low in the basement.

164. Pipe Sizes.—The selection of the proper size of pipes has been a much-debated question, and in the past there has been a wide variation in practice, particularly in the case of return pipes. The proper size of pipes is not always entirely a matter of calculation, for such things as dirt accumulations and faulty installation must be allowed for.

With steam piping there are two distinct phases to the problem. When the steam and condensate are flowing in the same direction, the pipe sizes may be chosen purely on a *pressure-loss* basis; that is, the pipes are selected of such sizes that the pressure drop caused by friction does not exceed a stipulated amount. This is true of supply mains, downfeed risers, and upfeed risers in two-pipe systems. When there is condensate flowing in the opposite direction, the permissible steam flow is governed not by the pressure drop but by limiting *velocity* of the steam beyond which the condensation will be picked up by the steam flow and prevented from draining properly. The capacities allowed for pipes with counterflowing condensation are less than when they are selected on a basis of moderate pressure drop.

In either case, in most buildings the velocities must not be so high as to create any noticeable noise; but there is no great danger of this if conservative limits of pressure drop are used.

The limiting or critical velocity with reverse flow of condensate will be discussed later. The matter of friction loss will be first considered.

165. Pressure Loss Due to Friction.—When any fluid flows through a pipe, a certain pressure is necessary to move it against the resistance caused by the friction of the fluid against the inner surface of the pipe. The following laws governing the friction of fluids in pipes have been established by experiment:

1. *The total amount of frictional resistance is independent of the absolute pressure of the fluid against the pipe wall.*

2. *The frictional resistance varies nearly as the square of the velocity.*

3. *The frictional resistance varies directly as the area of contact between the fluid and the pipe wall.*

4. *The frictional resistance varies directly as the density of the fluid.*

Consider a condition of steady flow in a pipe. Let p_1 (Fig. 134) be the unit static pressure of the fluid at one point, and let p_2 be the pressure at another point at a distance L from the first. The drop in pressure due to the friction of the fluid in passing through the distance L is then

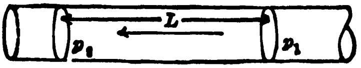

FIG. 134.

$$P = p_1 - p_2$$

Expressing the laws of friction stated above in algebraic form,

$$Pa = FSDv^2 \tag{1}$$

where P = drop in unit pressure, lb. per sq. ft.
a = cross-sectional area of the pipe, sq. ft.
F = a constant depending on the nature of the fluid and the nature of the pipe surface.
S = area of contact between the fluid and the pipe, sq. ft.
D = density of the fluid, lb. per cu. ft.
v = velocity of the flow, ft. per sec.

Then

$$P = \frac{1}{a} FSDv^2 \tag{2}$$

Let F be made arbitrarily $= \dfrac{f}{2g}$

Then Eq. (2) becomes

$$P = \frac{1}{a} fSD \frac{v^2}{2g} \tag{3}$$

This is done simply to bring into the expression the term $v^2/2g$, which is the usual form for expressions of this nature.

For round pipes of diameter d and length L, $S = \pi dL$, and $a = \pi d^2/4$. Then

$$P = \frac{4fLDv^2}{d2g} \tag{4}$$

Let w = the weight of steam flowing in lb. per min. Then

$$w = \frac{\pi d^2}{4} \times v \times D \times 60 = 47.12d^2vD$$

and

$$v = \frac{w}{47\ 12d^2D} \tag{5}$$

Let p be the pressure drop in lb. per sq. in. $= P/144$, and let d_1 be the diameter in inches $= 12d$.

Substituting in Eq. (4) these values for v, P, and d,

$$p = 0.04839\frac{fw^2L}{Dd_1^5} \tag{6}$$

The coefficient f was found by Unwin to be

$$f = K\left(1 + \frac{3}{10d}\right) = K\left(1 + \frac{3.6}{d_1}\right)$$

The value most commonly used for K for steam is that determined by Babcock, 0.0027.

Substituting in Eq. (6),

$$p = 0.0001306w^2L\frac{\left(1 + \frac{3.6}{d_1}\right)}{Dd_1^5} \tag{7}$$

where p = pressure drop, lb. per sq. in.
w = weight of steam flowing, lb. per min.
L = length of pipe, ft.
d_1 = diameter of pipe, in.
D = *average* density of steam, lb. per cu. ft.

The friction loss for many liquids and gases can be calculated by the use of an ingenious device known as the Reynolds number. To deal with it properly would require more space than the scope of this book permits. The student would profit by consulting the items in the bibliography at the end of this chapter relating to that subject.

166. Pressure-drop Chart.—Formula (7) is applicable to any steam pressure. The chart in Fig. 135 is based on an average

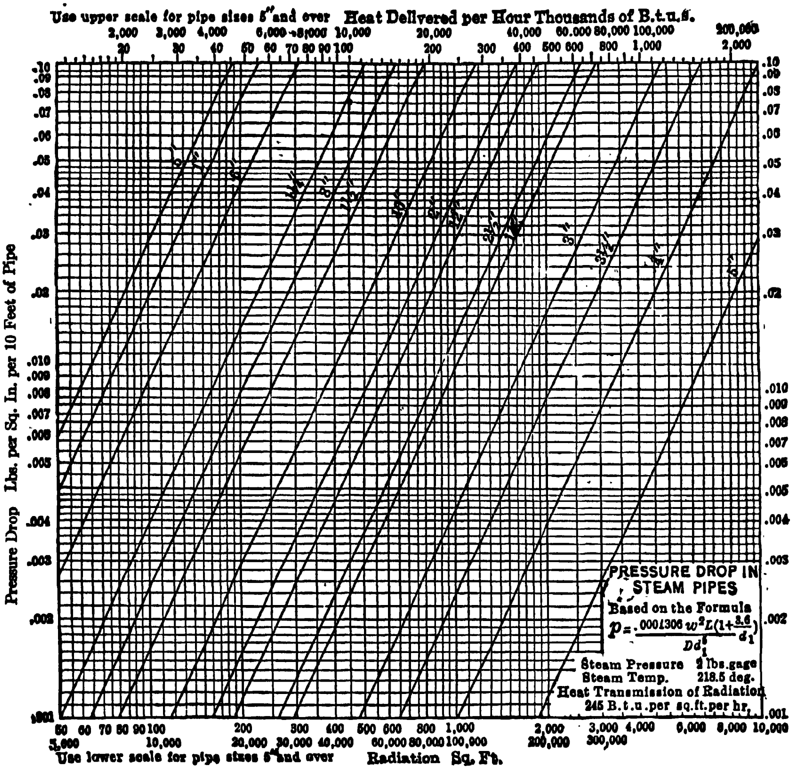

Fig. 135.—Pressure-drop chart.

steam pressure of 2 lb. per sq. in. gage and is convenient for heating work.

Example.—As an example of its use, assume that a pipe is to be designed for a total pressure drop of 2 lb. and that it is 500 ft. long and supplies 55,000 sq. ft. of radiator surface. What size of pipe is required?

The pressure drop is 0.04 lb. in 10 ft. Find the point on the horizontal scale corresponding to 55,000 sq. ft. of radiator surface, and follow vertically to intersect the diagonal line for a 10-in. pipe and for an 8-in. pipe. The 10-in. pipe would have a pressure drop of 0.022 lb. in 10 ft.; and the 8-in. pipe, 0.075 lb. in 10 ft., indicating that a 10-in. pipe is called for.

The design of pipes for the transmission of steam over considerable distances is usually based on the allowable pressure drop, and such problems are solved in the foregoing manner. When the pressure drop is beyond the range of the chart, or when the average steam pressure is considerably higher than 2 lb., it is necessary to use the formula.

The frictional resistance of the fittings must also be considered. It is convenient to reduce these resistances to equivalent lengths of straight pipe, to be added to the actual length, according to Table 48.

TABLE 48.—EQUIVALENT RESISTANCE OF FITTINGS

Fitting	Equivalent Length of Straight Pipe Expressed in Number of Pipe Diameters
90-deg. elbow	40
45-deg. elbow	20
Tee	40
Reducing coupling	40
Globe valve	60

167. The Counterflow of Condensation.—The pressure-drop method is suitable when the condensation flows in the direction of the steam; but in upflow risers of a heating system, especially in a single-pipe system, the condensation must drain back against the steam flow. Laboratory studies[1] have thrown considerable light on this subject.

At low velocities the counterflowing condensation runs down the inner surface of the pipe in smooth and even streams. The steam flows up the center of the pipe, and there is no interference between the downcoming condensate and the upgoing steam. As the velocity increases and more steam flows up the pipe with a corresponding increase in the amount of returning condensate, there is some interference between the two streams until a critical velocity is reached at which the condensate, which has accumulated in the pipe, is suddenly carried up into the radiator. In a single-pipe system, where there is no other outlet for the condensate, it periodically flows back in slugs against the steam flow or accumulates in the radiator and causes noisy disturbances.

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[1] These tests, made at the A.S.H. & V.E. laboratory, are reported in various papers in the *Transactions* of that society for 1922 and later.

The entrance to the vertical riser, that is, the outlet of the tee or elbow at the bottom, is the point at which the picking up of the condensate originates, and what takes place is illustrated in Figs. 136 and 137. The former shows how the first interference is caused by the drops of condensate being caught by the steam and shot up into the riser. Figure 137 shows how the rings or waves of condensate form at the entrance and restrict the steam passage, increasing the steam velocity at that point until a further increase in flow causes the picking up of the entire mass of water to take place.

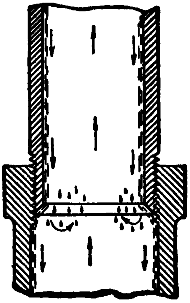

FIG. 136.—Drops of condensate, caught by steam.[1]

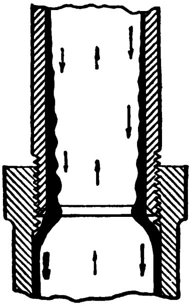

FIG. 137.—Ring of condensate, formed at entrance of riser.[1]

The same sort of action would naturally occur at any point of restriction, and it is therefore evident that the capacity of a riser with counterflowing condensate is limited by the most restricted area. A radiator valve with restricted opening or a pipe that is dented or not properly reamed may have a serious effect. The critical velocity is lower in horizontal or inclined pipes than in vertical pipes and horizontal offsets in risers; therefore branches to radiators, especially in a one-pipe system, should be relatively larger than vertical risers.

The relation of pressure drop in risers to steam flow as shown by laboratory tests[2] is illustrated in Fig. 138. The curves flatten out at certain rates of flow, indicating that the critical velocity and maximum capacity have been reached and that the condensate is being picked up.

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[1] From HOUGHTEN, EBIN, and LINCOLN, "Critical Velocity of Steam and Condensate Mixtures in Horizontal, Vertical, and Inclined Pipes," *Trans. A.S.H. & V.E.*, 1924.

[2] HOUGHTEN and O'CONNELL, "Capacity of Up-feed Steam Heating Risers for One- and Two-pipe Systems, *Trans. A.S.H. & V.E.*, 1927

168. Practical Selection of Pipe Sizes.—The capacity of upfeed risers in one-pipe systems is thus limited by *interference of the condensate.* In practice the capacities assumed for various pipe sizes are below those determined by laboratory test, to take care of possible defective or partially clogged piping.

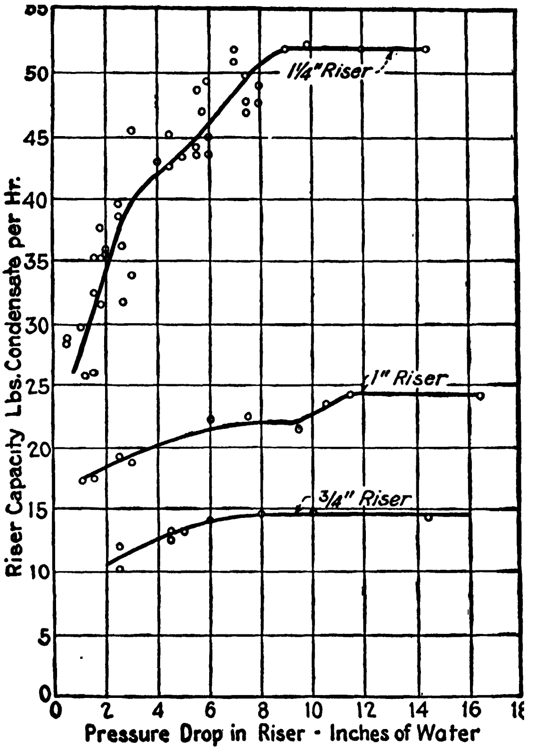

FIG. 138.—Relation between pressure drop and steam flow in risers in a single-pipe system.

The capacities of mains and downfeed risers in all systems, and of upfeed risers in *two-pipe* systems, are based on considerations of *pressure drop.* It is true that with upfeed risers in two-pipe systems there is a small amount of counterflowing condensate that may be picked up by the steam, but the rate of flow at which this occurs is relatively high and is not reached in pipes designed on the basis of moderate pressure drop.

In studying the practical selection of pipe sizes, the first case to be considered is one in which some particular drop through the system is desired. The pipe sizes to give that pressure drop will be selected from the chart in Fig. 135. Later a table will be given in which the pipe capacities for certain pressure drops are shown.

Consider the overhead vapor system shown diagrammatically in Fig. 139, and let it be required to choose the pipe sizes so that the pressure drop through the system will be between 0.3 and 0.5 lb. The equivalent lengths of the sections of pipe should first be computed and set down in tabular form. Obviously, since the pipe sizes are not yet known, it will be necessary to assume them in order to compute the allowance for the fittings as given in Table 48. In a highly refined calculation it would be necessary to refigure the equivalent lengths after the first trial computation of pipe sizes, but for practical purposes the first assumption is close enough. After ascertaining that the longest steam flow path is from the boiler, up the main riser, and down rise *hp*, it would next be essential to determine the pressure drop through each section of the piping, assuming an initial pressure of 2 lb. at the boiler. The total length of the path being 387 ft., the average pressure drop may be taken as 0.4 ÷ 38.7 = 0.010 lb. per 10 ft. of pipe. The pressure drop through each of the successive sections may then be computed from the chart in Fig. 135, using the pipe sizes that will give as nearly as possible the

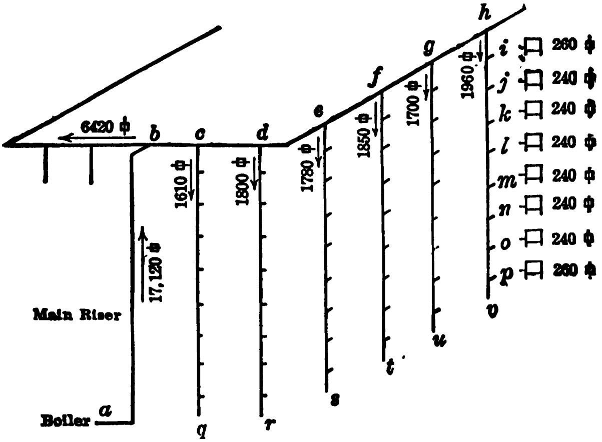

FIG. 139.

average pressure drop determined above. The results may be arranged in tabular form as in Table 49.

In systems of this kind it is desirable to have about the same pressure at all of the lowest radiators. The other risers therefore can be designed for such a pressure drop that the pressure at the bottom of each will be approximately 1.556 lb.

169. Pipe-size Table.—For the practical selection of pipe sizes it is usually sufficiently accurate to use Tables 50 and 51, which are based on research work conducted by the A.S.H. & V.E. Table 50 gives the capacities of pipes carrying steam. Columns *B* to *G* are based on different rates of pressure drop, and it is advisable to choose the over-all pressure drop through the system and divide that figure by the equivalent length of the longest circuit to obtain the pressure gradient, which will then show the proper column in Table 50 to use.

TABLE 49

Section	Equivalent length, ft.	Rad. supplied, sq. ft.	Initial pressure, lb.	Pipe size, in.	Pressure drop in section, lb.
a–b	130	17,120	2.000	8	0.0075 × 13.0 = 0.0974
b–c	20	10,700	1.903	6	0.0125 × 2.0 = 0.0250
c–d	23	9,090	1.878	6	0.0090 × 2.3 = 0.0210
d–e	19	7,290	1.857	5	0.0150 × 1.9 = 0.0290
e–f	27	5,510	1.828	5	0.0090 × 2.7 = 0.0240
f–g	23	3,660	1.804	4	0.0130 × 2.3 = 0.0300
g–h	25	1,960	1.774	3	0.0170 × 2.5 = 0.0420
h–i	15	1,960	1.732	3	0.0170 × 1.5 = 0.0250
i–j	15	1,700	1.707	3	0.0130 × 1.5 = 0.0190
j–k	15	1,460	1.688	3	0.0090 × 1.5 = 0.0140
k–l	15	1,220	1.674	2½	0.0220 × 1.5 = 0.0330
l–m	15	980	1.641	2½	0.0140 × 1.5 = 0.0210
m–n	15	740	1.620	2	0.0220 × 1.5 = 0.0330
n–o	15	500	1.587	2	0.0100 × 1.5 = 0.0150
o–p	15	260	1.572	1½	0.0110 × 1.5 = 0.0160
	387				

NOTE.—Final pressure at *p*, 1.556 lb.; total drop, 0.444 lb.

The total pressure drop from the boiler to the farthest radiator should not be greater than one-half of the assumed pressure at the boiler. For large vacuum systems a total pressure drop of ¼ to ½ lb. to the farthest radiator is good practice. For small

TABLE 50.—STEAM-PIPE CAPACITIES[1] IN SQUARE FEET OF EQUIVALENT RADIATION
(Reference to this table will be by column letter *A* through *L*)

Pipe size, in.	Capacities of steam mains and risers								Special capacities for one-pipe systems only		
	Direction of condensation flow in pipe line										
	With the steam in one-pipe and two-pipe						Against the steam two-pipe only		Supply risers upfeed	Radiator valves and vertical connections	Radiator and riser runouts
	1/32-lb. drop	1/24-lb. drop	1/16-lb. drop	1/8-lb. drop	1/4-lb. drop	1/2-lb. drop	Vertical	Horizontal			
A	*B*	*C*	*D*	*E*	*F*	*G*	*H**	*I*†	*J*‡	*K*	*L*†
¾	......	......	30	....	....	....	30	.....	25		
1	39	46	56	79	111	157	56	26	45	20	20
1¼	87	100	122	173	245	346	122	58	98	55	55
1½	134	155	190	269	380	538	190	95	152	81	81
2	273	315	386	546	771	1,091	386	195	288	165	165
2½	449	518	635	898	1,270	1,797	635	395	464	...	260
3	822	948	1,163	1,645	2,326	3,289	1,129	700	799	...	475
3½	1,228	1,419	1,737	2,457	3,474	4,913	1,548	1,150	1,144	...	745
4	1,738	2,011	2,457	3,475	4,914	6,950	2,042	1,700	1,520	...	1,110
5	3,214	3,712	4,546	6,929	9,092	12,858	.....	3,150	.....	...	2,180
6	5,276	6,094	7,642	10,553	14,924	21.105					
8	10,983	12,682	15,533	21,967	31,066	43,934					
10	20,043	23,144	28,345	40,085	56,689	80,171					
12	32,168	37,145	45,492	64,336	90,985	128,672					
16	60,506	69,671	84,849	121,012	169,879	240,245					
	All horizontal mains and downfeed risers						Upfeed risers	Mains and undripped runouts	Upfeed risers	Radiator connections	Runouts not dripped

[1] This table is based on pipe-size data developed through the research investigations of the A.S.H. & V.E. Reprinted by permission.

NOTE.—All drops shown are in pounds per 100 ft. of equivalent run—based on pipe properly reamed.

* Do not use column *H* for drops of 1/24 or 1/32 lb.; substitute column *C* or column *B* as required.

† On radiator runouts over 8 ft. long increase one pipe size over that shown in Table 50.

‡ Do not use column *J* for drop of 1/32 lb. except on sizes 3 in. and over; below 3 in. substitute column *B*.

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TABLE 51.—RETURN-PIPE CAPACITIES[1] IN SQUARE FEET OF EQUIVALENT RADIATION

(Reference to this table will be by column letter *M* through *EE*)

Pipe size, in.	Capacity of return mains and risers																	
	Mains																	
	1/32-lb. drop per 100 ft.			1/24-lb. drop per 100 ft.			1/16-lb. drop per 100 ft.			1/8-lb. drop per 100 ft.			1/4-lb. drop per 100 ft.			1/2-lb. drop per 100 ft.		
	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.
M	*N*	*O*	*P*	*Q*	*R*	*S*	*T*	*U*	*V*	*W*	*X*	*Y*	*Z*	*AA*	*BB*	*CC*	*DD*	*EE*
3/4	......	......	.....	......	......	326	......	......	400	......	......	568	......	......	800	....	....	1,130
1	500	248	.....	580	285	570	700	320	700	1,000	412	994	1,400	460	1,400	....	....	1,977
1 1/4	850	520	.....	990	595	976	1,200	670	1,200	1,700	868	1,704	2,400	962	2,400	....	....	3,390
1 1/2	1,350	822	.....	1,570	943	1,547	1,900	1,058	1,900	2,700	1,362	2,696	3,800	1,512	3,800	....	....	5,370
2	2,800	1,880	.....	3,240	2,140	3,256	4,000	2,300	4,000	5,600	2,960	5,680	8,000	3,300	8,000	....	....	11,300
2 1/2	4,700	3,040	.....	5,300	3,470	5,453	6,700	3,800	6,700	9,400	4,900	9,510	13,400	5,450	13,400	....	....	18,925
3	7,500	5,840	.....	8,500	6,250	8,710	10,700	7,000	10,700	15,000	9,000	15,190	21 400	10,000	21,400	....	....	30,230
3 1/2	11,000	7,880	.....	13,200	8,800	13,020	16,000	10,000	16,000	22,000	12,900	22,710	32,000	14,300	32,000	....	....	45,200
4	15,500	11,700	.....	18,300	13,400	17,910	22,000	15,000	22,000	31,000	19,300	31,220	44,000	21,500	44,000	....	....	62,180
5	......	......	.....	......	......	31,500	......	......	38,700	......	......	54,920	......	......	77,400	....	....	109,300
6	......	......	.....	......	......	50,450	......	......	62,000	......	......	88,000	......	......	124,000	....	....	175,100
	Risers																	
3/4	......	190	.....	......	190	570	......	190	700	......	190	994	......	190	1,400	....	....	1,977
1	......	450	.....	......	450	976	......	450	1,200	......	450	1,704	......	450	2,400	....	....	3,390
1 1/4	......	990	.....	......	990	1,547	......	990	1,900	......	990	2,696	......	990	3,800	....	....	5,370
1 1/2	......	1,500	.....	......	1,500	3,256	......	1,500	4,000	......	1,500	5,680	......	1,500	8,000	....	....	11,300
2	......	3,000	.....	......	3,000	5,453	......	3,000	6,700	......	3,000	9,510	......	3,000	13,400	....	....	18,925
2 1/2	......	......	.....	......	......	8,710	......	......	10,700	......	......	15,910	......	......	21,400	....	....	30,230
3	......	......	.....	......	......	13,020	......	......	16,000	......	......	22,710	......	......	32,000	....	....	45,200
3 1/2	......	......	.....	......	......	17,910	......	......	22,000	......	......	31,220	......	......	44,000	....	....	62,180
4	......	......	.....	......	......	31,500	......	......	38,700	......	......	54,920	......	......	77,400	....	....	109,300
5	......	......	.....	......	......	50,450	......	......	62,000	......	......	88,000	......	......	124,000	....	....	175,000

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[1] This table is based on pipe-size data developed through the research investigations of the A.S.H. & V.E. Reprinted by permission.

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systems of all types, where the equivalent length is less than 200 ft. to the farthest radiator, a pressure drop of 1/16 lb. per 100 ft. is about right, and column *D* of Table 50 is therefore used. It will be noted that in Table 50 a distinction is made between cases where the condensation flow is with the steam or against the steam. The reason for this should be evident from the previous consideration of the effect of the condensation on the steam flow, and an analysis of the conditions existing in each section of the piping will readily indicate which column should be used. The table is based on the assumption that all the pipe ends are carefully reamed.

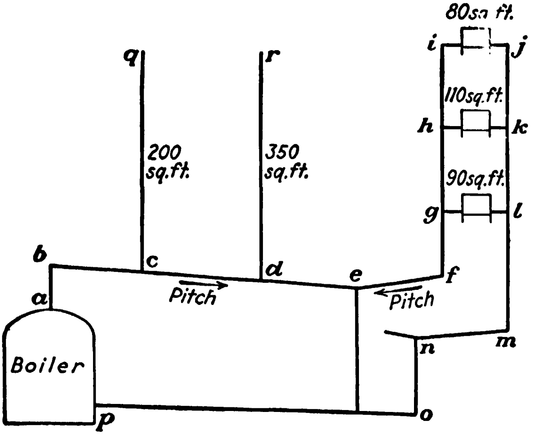

FIG. 140.

Table 51 shows return pipe sizes for wet, dry, and vacuum returns. The same rate of pressure drop should be used as is assumed for the steam main of the system.

Example.—To select from the tables the pipe sizes for the two-pipe gravity system shown in Fig. 140. (The return pipes and radiators are shown for only a portion of the system.) The individual radiator sizes for riser *fi* and the total radiator surface for risers *dr* and *cq* are indicated.

The lengths of pipe in the respective sections are given in Table 52, and from Fig. 140 the radiator surface served by each section of steam and return pipe can readily be calculated. Since the equivalent length of the circuit *ai* is considerably less than 200 ft., a pressure-drop rate of 1/16 lb. per 100 ft. will be used and the sizes of the sections between *a* and *e* will be taken from column *D*, Table 50. Table 52 also indicates the columns in Tables 50 and 51 from which the pipe size for each section is selected. The last column in Table 52 gives the pipe sizes.

TABLE 52

Section	Radiator surface supplied, sq. ft.	Length of pipe, ft.	Column to be used in tables	Pipe size, in., from table
ab	830	5	*D*	3
bc	830	20	*D*	3
cd	630	22	*D*	2½
de	280	12	*D*	2
ef	280	10	*I*	2½
fg	280	4	*H*	2
gh	190	10	*H*	1½
hi	80	10	*H*	1¼
ik	80	10	*U*	1
kl	190	10	*U*	1
lm	280	6	*U*	1
mn	280	12	*U*	1
no	830	6	*U*	1½
op	830	60	*T*	1¼

In a large system where it is necessary to assume the equivalent length of pipe before the pipe sizes can be chosen, a safe assumption is to take the equivalent length of pipe as twice the actual length, checking back later if necessary.

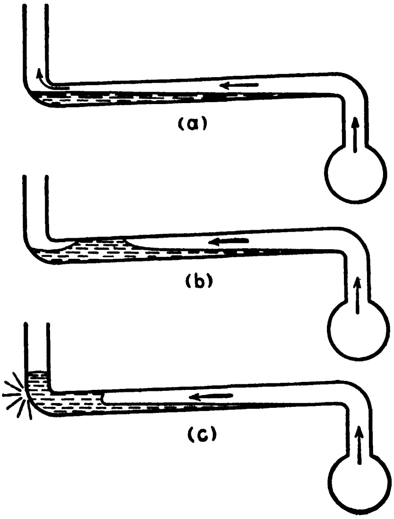

FIG. 141.—Water hammer in steam pipe.

170. Noise in Piping Systems.—There are two kinds of noise in steam heating systems. The more common and the more objectionable is the well-known water hammer, resembling a tapping or pounding in the pipes. It is caused by improper drainage of the condensation from the steam pipes, which in turn is a result of faulty installation or of failure of a drainage trap to function. The noise is caused by slugs or pistons of water being driven against a dead end or sharp turn in the pipe. The forces set up are of great magnitude and frequently result in the breaking of cast-iron fittings.

A common cause of water hammer is illustrated in Fig. 141, which shows what probably happens in a branch to a riser that is incorrectly pitched so as to form a water pocket. In Fig. 141*a* the pipe is shown partially full of water but with sufficient space above to permit the steam to pass. As the amount of water increases, owing to condensation, the velocity of the steam through the restricted area becomes so great that a wave is picked up by the steam, as shown at *b*, and is driven against the side of the elbow with great force, as shown in *c*. Some of the water is carried up the riser, or perhaps the piston of water breaks up, and the cycle is repeated.

The precautions to prevent water hammer are obviously the proper pitching of pipes, the avoidance of undrained pockets, and the choosing of the proper sizes of pipe so as to avoid high steam velocities when there is a counterflow of condensate. The pipe-size tables, if carefully followed, are liberal enough to make the velocities safely low.

The other type of noise is a hissing sound due to high steam velocities. It may occur in any part of the heating system when steam is first turned on but not under normal conditions unless the pipes are far undersized. The velocity at which noise begins varies with the pitch of the pipes and with the amount of condensate flowing.[1]

Radiator orifices may cause noise if they are so small as to require large pressure drops but will not do so if the sizes given in Chap. XI are used.

Pressure-reducing valves, through which steam necessarily flows at high velocities, are difficult to make quiet, although much can be done by covering them with felt or another soft material.

171. Erection and Installation of Piping.—It is very necessary that the installation of a heating system be supervised carefully, as an immense amount of trouble can be caused by careless workmanship.

One of the most important points is the proper threading and making up of the pipe joints. Sharp, clean threads of the proper length should be the aim, the cutting of which requires that the threading dies be kept in perfect condition. In making up the joints, the threads should be wiped perfectly clean and coated with a very small amount of pipe-joint compound. The use of too great a quantity of compound is a frequent and serious mistake, as the substance clogs the traps, valves, and return lines and is a continual source of trouble.

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[1] See HOUGHTEN and others, "Capacity of Radiator Supply Branches for One- and Two-pipe Systems," *Trans. A.S.H. & V.E.*, 1929.

Pipes of the 3-in. size and under are cut with a hand cutter, which leaves a burr on the inside of the pipe. In the smaller pipes especially, a considerable reduction in the internal diameter may thus be produced, and the burr should therefore be removed with a reamer.

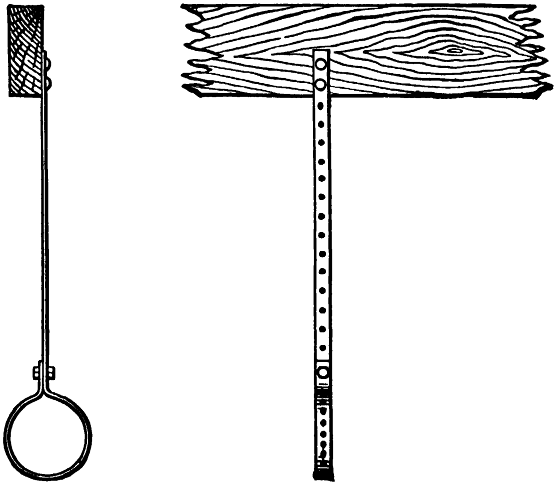

Fig. 142.—Simple form of pipe hanger.

The piping should be uniformly pitched, and all air or water pockets should be avoided. Hangers should be installed in sufficient numbers and in proper locations so that no strains on fittings, valves, or boiler connections will be caused by the weight of the piping.

One common source of trouble, especially in new installations, is the dirt that gets into the piping while it is being installed. This dirt, consisting of cement, plaster, chips, etc., from the building operations, and chips produced in threading the pipe, causes a great deal of damage in clogging the pipes, traps, and fittings and in scoring the valve seats and disks. The most important precaution is to keep the open ends of the piping carefully covered during installation, to prevent dirt from entering.

Systems having traps on the radiators should be operated for a week or two without the traps so that most of the dirt will be washed out before the traps are installed.

172. Pipe Hangers.—The piping in a heating system must be substantially supported either from the building structure or from special supports. Horizontal mains are usually hung from the joists or steelwork of the floor above. For pipes of moderate size the hanger shown in Fig. 142 is widely used. The perforated metal can be obtained in strips and cut to any required length. This hanger is of low cost and can be installed very cheaply.

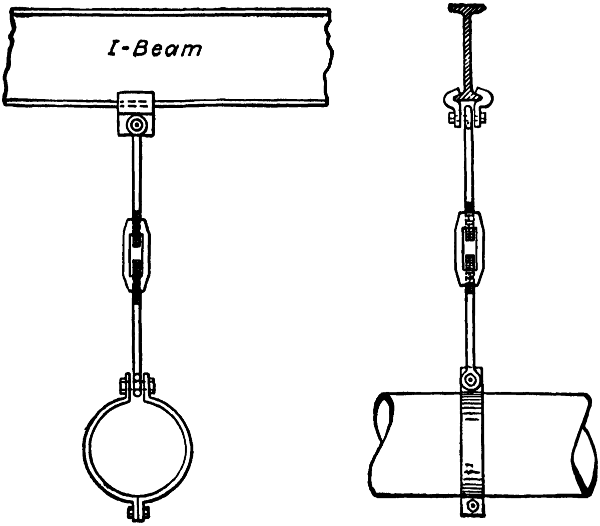

FIG. 143.—Hanger for large pipes.

For heavier pipes the hanger shown in Fig. 143 is a common design. The turnbuckle is used to adjust the elevation of the pipe when it is being installed. Both of these hangers permit the free longitudinal movement of the pipe line. Hangers should be placed at intervals of 20 ft. or less on all horizontal pipes.

173. Boiler Connections.—The most important precaution to be taken in arranging the piping connections to the boiler is in regard to the return connections. In a heating system there is always a possibility that an unbalanced pressure condition will cause the water to be forced out of the boiler and back into the return line, with consequent damage to the boiler through overheating. Also, when two or more boilers are installed, any difference in frictional resistance in the outlet steam piping tends to cause unequal water levels in the boilers.

With a single boiler a check valve in the return line is a common precaution and in some cities is required by law. An alternate method, which is more effective, is the Hartford Loop, shown in Fig. 144. The return line is brought up to the water level of the boiler and connected into a vertical balance connection, which ties into both the steam-outlet pipe and the return opening.

The recommended size for the Pipe C is as follows:

Grate Area	*Size of Pipe C*
4 square feet or less	*1½ in.*
4 square feet to 15 sq. ft.	*2½ in.*
15 square feet or more	*4 in*

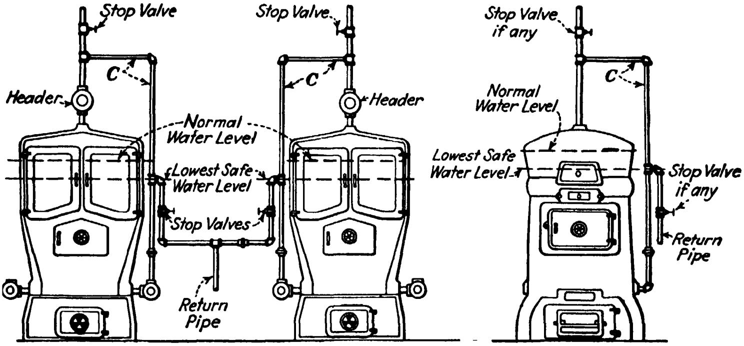

Fig. 144.—Hartford return connection.[1]

Where two or more boilers are installed, this form of connection is much better than check valves and is highly desirable.

174. Piping Drawings.—Certain conventional methods have come into use for representing heating and ventilating systems. Symbols are used to advantage to represent common kinds of equipment, in order to conserve drafting-room time and to simplify drawings. A symbol should resemble approximately the apparatus that it represents, and a code of symbols should be reproduced on the drawing. The standard symbols of the A.S.A. are shown in the Appendix.

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[1] Reproduced by permission of the Hartford Steam Boiler Inspection and Insurance Company.

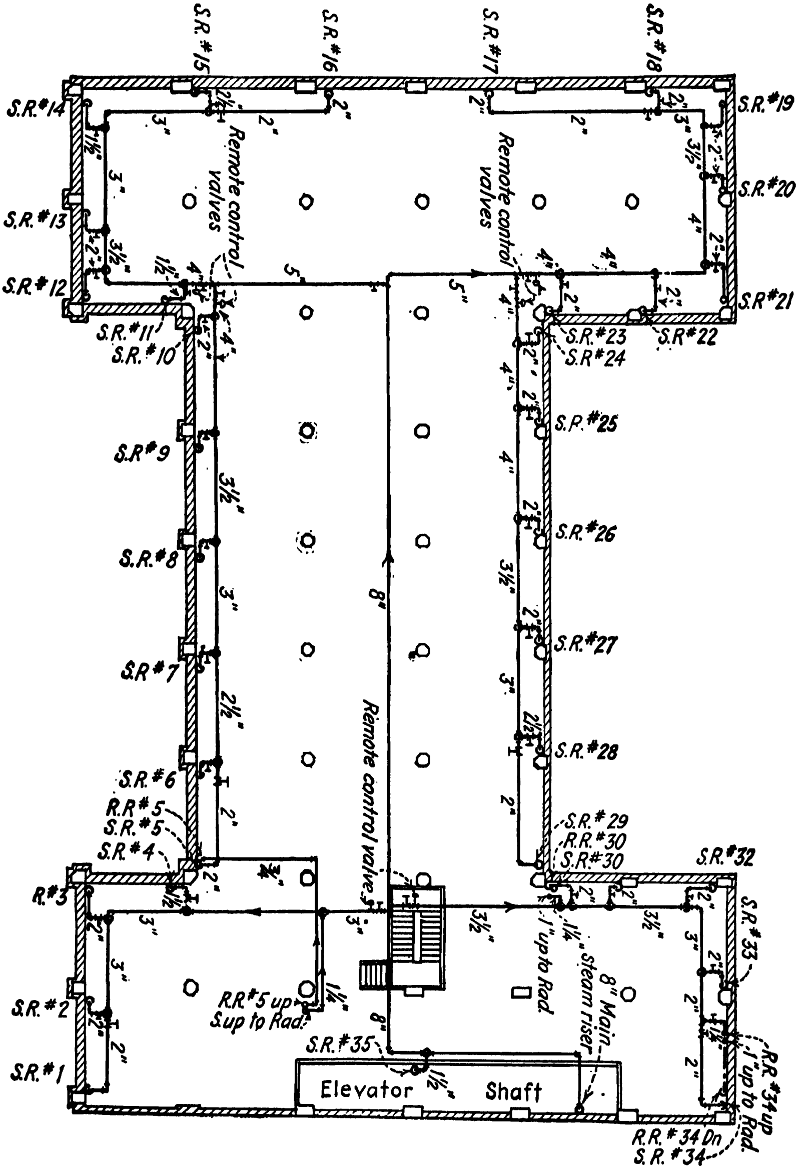

Fig. 145.—Plan of overhead supply main.

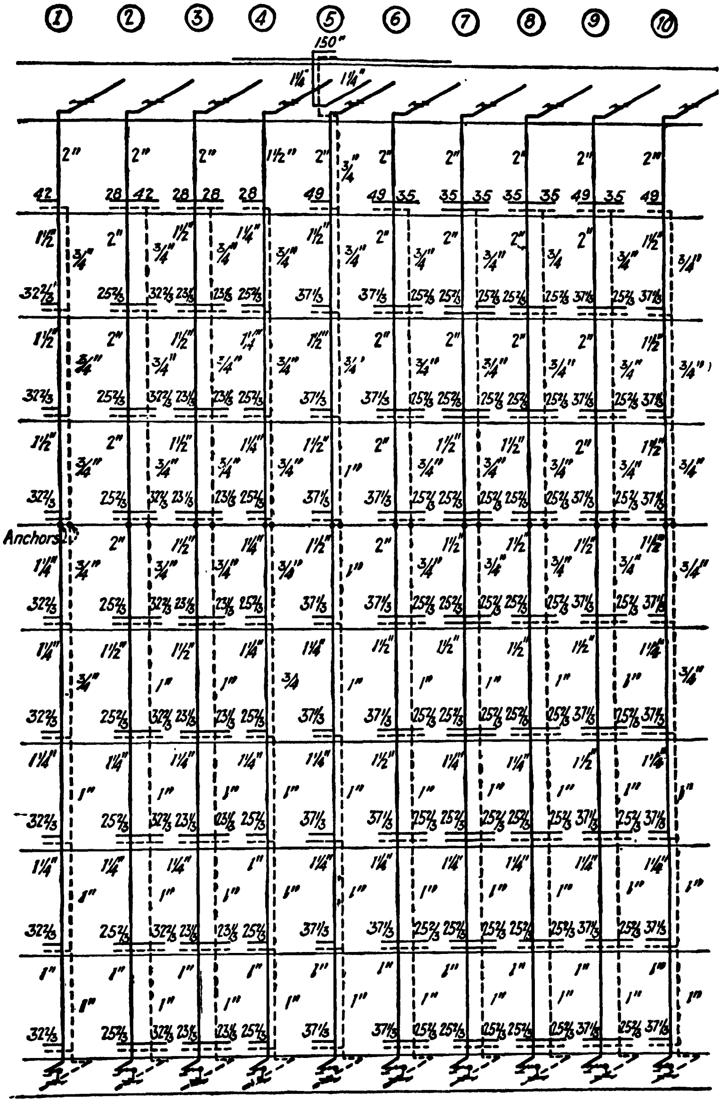

Fig. 146.—Riser diagram.

Figure 145 shows a plan of an overhead supply main in a downfeed system, and Fig. 146 shows a riser diagram for the same building. The riser diagram is an important drawing to the contractor installing the heating system, and one should be made for all except very small systems. It shows the risers designated by number, the pipe sizes, and the radiator sizes.

175. Troubles in Steam Heating Systems.—The following is a brief discussion of some of the more common troubles encountered in steam heating systems.[1] Practical experience is the best teacher in this respect. The diagnosis usually calls for a thoughtful analysis of the case, preceded by a thorough inspection and tracing out of all of the piping that might be involved.

As to boiler troubles, **low water** is a frequent difficulty. This may be caused by frozen or plugged return lines, leaks (especially in buried portions), insufficient elevation of first-floor radiators above the boiler in a vapor system, or priming. The remedies in most cases are obvious.

Priming is usually caused by impurities in the water, and an **unsteady water line** is another manifestation of it. Sometimes draining the boiler and refilling will suffice, but often the trouble is due to oil, and the boiler should be given a generous dose of sodium carbonate (soda ash), and a light fire should be carried for some time, followed by a thorough blowdown. A surface blowoff through the safety-valve opening is often resorted to.

When two boilers are operated in parallel, the water lines are apt to be unstable unless the piping connections are absolutely symmetrical or unless the Hartford Loop (Fig. 144) is used.

In the category of piping troubles, **noise** is a frequent complaint. Pounding, gurgling, or tapping noises are caused by accumulations of water, usually because of improper pitch of the pipes or defective drainage. Drip traps should be inspected. Creaking noises are usually caused by interferences with the expansion of the pipes by contact with the building structure.

Insufficient heat may be due to many causes. If it is general throughout the building, the first thing to be sure of is that there is a good hot fire, which may not be the case if there is insufficient draft. If there is plenty of steam pressure in all radiators, the radiators may be insufficient, or there may be excessive infiltration into the building.

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[1] See also ALFORD G. CANAR, "Check List for Heating Servicing," *Heating & Ventilating*, October, 1938.

Failure of radiators to heat is often due to air binding, and the air vents on risers and on radiators should be inspected. Radiators sometimes become loaded with water because of plugged traps or return pipes.

Corrosion rarely becomes serious in steam heating systems under normal water conditions. In large plants where make-up water is used, the return pipes may become corroded by the action of oxygen or carbon dioxide. A survey should be made by a competent chemist before any remedies are attempted.

Ammonia, sometimes present in water supplies, causes corrosion of brass and bronze parts.

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Problems

1. Compute the increase in length of a steel steam pipe 87 ft. in length when filled with steam at 10 lb. pressure, if the pipe was originally at a temperature of 60°.

2. Compute the increase in length of a steel steam pipe 217 ft. in length when filled with steam at 125 lb. pressure. Original temperature 60°.

3. How much steam can be transmitted by a 6-in. pipe 93 ft. long with an initial pressure of 5 lb. gage and a final pressure of 4 lb. gage?

4. How much steam can be transmitted by the same pipe as in Prob. 3 with an initial pressure of 105 lb. gage and a final pressure of 104 lb. gage?

5. What will be the drop in pressure if 2,000 lb. of steam per hr. at an initial pressure of 10 lb. gage is passed through a 5-in. pipe 87 ft. long containing three 90-deg. elbows?

6. What initial pressure will be required if 110 lb. of steam per min. flows through a 4-in. pipe 70 ft. long, the final pressure being 51 lb. gage? The pipe has two 90-deg. elbows.

7. By means of the method of Art. 168, compute the pipe sizes for the heating system of Fig. 139 with a pressure drop through the system of approximately 1 lb.

8. Select the pipe sizes from point *b* to point *p* in the system of Fig. 139, using Table 50 and assuming an over-all pressure drop from *b* to *p* of ½ lb.

9. Select the pipe sizes for a return riser for the steam riser *h*-*v* (Fig. 139) and a return main from *v* to *a*, picking up the other return risers. Assume the system to be a vacuum system.

CHAPTER X

HOT-WATER HEATING SYSTEMS[1]

176. Types of Systems.—The hot-water heating system utilizes water as the means of conveying the heat from the boiler to the radiators. The design of a hot-water system, after the radiators and boiler have been selected according to the methods outlined in previous chapters, consists in laying out the piping system and selecting the proper pipe sizes. Both of these operations must be carefully done, but the principles involved are simple.

The circulation in a hot-water system may be produced simply by the difference in density of the hot water in the pipes leading to the radiators and the cooler water in the pipes returning from the radiators, or it may be produced by a pump. The former method requires much larger pipes, because the force available to produce gravity circulation is quite small, whereas the forced-circulation method requires relatively small pipe sizes. The present trend is distinctly toward the forced-circulation method.

There are numerous piping arrangements that may be used, but the five commonest types are shown diagrammatically in Fig. 147. The two-pipe system shown in Fig. 147*a* is an arrangement for residence systems and other buildings of small or moderate size. The supply and return mains circle the basement, and water flows from the supply main through one radiator and into the return main.

The system of Fig. 147*b* is similar except that the return main is reversed so that the length of the path of flow through every radiator is approximately the same, tending to equalize the flow.

In Fig. 147*c* is shown a one-pipe system in which a single main only is used. The return pipe from each radiator is connected back into the main at a point downstream from the supply connection. In a gravity system arranged in this way the temperature of the water supplied to each radiator along the main is successively lower, and the size of the radiators must therefore be correspondingly increased. With forced circulation, provided the system is divided into enough circuits so that there are not many radiators on each circuit, it is not usually necessary to increase downstream radiator sizes. The overhead systems of Fig. 147*d* and 147*e* are used in large buildings.

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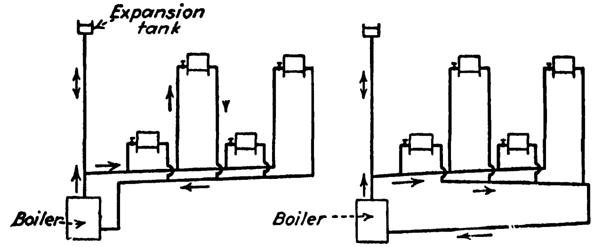
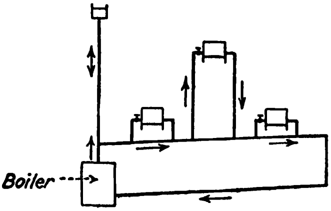
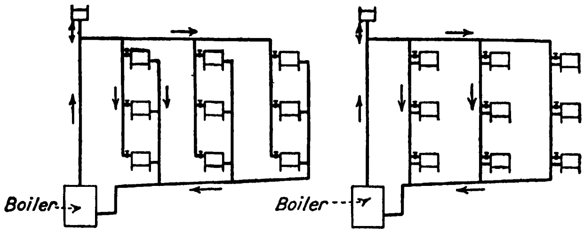
[1] The tables and charts and much of the material in this chapter were prepared by F. E. Giesecke, Ph.D., director, Texas Engineering Experiment Station, Agricultural and Mechanical College of Texas.



(a)-Two-Pipe System (b)-Two-Pipe System with Reversed Return



(c)-One-Pipe System



(d)-Overhead Two-Pipe System (e)-Overhead One-Pipe System

FIG. 147.—Diagrammatic arrangement of hot-water heating systems.

In general, the system of Fig. 147*a* or 147*b* is used for small buildings with gravity circulation; that of Fig. 147*c*, in small buildings with forced circulation; and that of Fig. 147*d*, in large buildings with either method of circulation.

When forced circulation is used, the pump is ordinarily placed in the circuit at a point just before the water enters the boiler or heater.

In each of these systems an expansion tank is shown, the function of which is to accommodate the change in volume of the water in the system under varying temperatures.

177. Theory of Flow in a Gravity System.—The force that produces flow in the gravity system is due to the difference in density of the hot water in the supply pipes and the cooler water in the return pipes. The conditions that exist after the flow has reached its maximum and constant velocity are the important consideration, because the system is to be designed for these conditions. This principle is invariably true—that **the force creating circulation is entirely expended as friction;** thus a circulation rate is obtained that causes the friction just to equal the circulating force.

Therefore the process of designing a gravity hot-water system consists principally of arranging the piping and selecting pipe sizes in which the friction loss, for that quantity of water necessary to convey the required amount of heat, will be approximately equal to the force available for producing circulation.

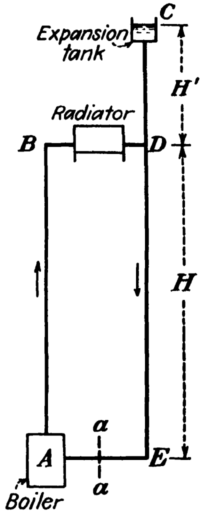


FIG. 148.

In Fig. 148 is shown an elementary system in which the radiator is located at a height H above the boiler, and the water level in the expansion tank is at a height H' above the radiator. Assume that the water in the flow riser AB is at temperature t_f and has a corresponding density D_f and that the water in the return riser DE has a lower temperature t_r and a correspondingly greater density D_r. The water in the riser CD to the expansion tank has a density D'.

Assume that the flow of water is prevented by an obstruction at the plane aa, such as a closed valve, and consider the forces acting against plane aa, which tend to produce flow. The pressure on the right-hand side is due to the water head in the pipe DE plus the head in the pipe CD, or

$$P_1 = HD_r + H'D' \tag{1}$$

If H and H' are expressed in feet, and D_r and D' in pounds per cubic foot, then P_1 will be the pressure in pounds per square foot.

The pressure on the left-hand side of *aa* is

$$P_2 = HD_f + H'D' \tag{2}$$

and the difference, tending to produce flow, is

$$p = H(D_r - D_f) \tag{3}$$

The density of water at various temperatures is given in Table 58,[1] and from these data it is possible to calculate p in pounds per square foot, knowing the value of h and the temperatures t_f and t_r. It is more convenient, however, to use as a unit of pressure the milinch head of water, which is the pressure exerted by a column of water one-thousandth of an inch in height and of a density equal to the mean of D_r and D_f. In symbols,

$$p = \frac{h}{12{,}000} \times \frac{D_r + D_f}{2} \tag{4}$$

where h = pressure head, milinches of water.
$h/12{,}000$ = pressure head, *feet* of water.
As before,
p = pressure difference, lb. per sq. ft.

Then

$$p = \frac{h}{12{,}000} \times \frac{D_r + D_f}{2} = H(D_r - D_f) \tag{5}$$

$$h = 24{,}000H\frac{D_r - D_f}{D_r + D_f} \tag{6}$$

The chart in Fig. 158 gives the pressure head h for different flow and return temperatures for 1 ft. of water column H. The head tending to produce circulation can thus be readily determined from the flow and return temperatures, and this is the first step in the design of a gravity system.

178. Friction Heads.—The frictional resistance to the flow of water depends upon the velocity of flow and the pipe diameter, and to a lesser extent upon the temperature of the water and the degree of roughness of the pipe surface. The friction head in a pipe varies as the 1.78 power of the velocity, and the friction head in a pipe fitting varies as the square of the velocity. The friction head in an elbow can therefore be expressed in terms of the friction head of an equivalent length of pipe.

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[1] The reference tables and charts in this chapter are placed at the end of the chapter.

Since the friction heads of all fittings vary practically as the square of the velocity, it is permissible to express the friction heads of the several types of fittings, valves, etc., in terms of the friction heads of an ordinary elbow.

Careful determinations have been made of the friction of water in pipes and fittings, with the results expressed in mathematical formulas, tables, and diagrams. Table 53 shows the friction heads of various fittings in terms of that of one standard elbow.

TABLE 53.—ELBOW EQUIVALENTS

	For steel pipe	For copper pipe
90-deg. elbow	1.0	1.0
45-deg. elbow	0.7	0.7
90-deg. long-radius elbow	0.5	0.5
Open return bend	1.0	1.0
Open gate valve	0.5	0.7
Open globe valve	12.0	17.0
Angle radiator valve	2.0	3.0
Radiator	3.0	4.0
Boiler or heater	3.0	4.0
Tee, per cent flowing through branch:		
10	100.0	100.0
25	14.0	20.0
50	4.0	4.0
75	2.3	1.8
100	1.8	1.2

NOTE.—These relations are very nearly correct for the low velocities obtaining in gravity circulation; for the higher velocities employed in forced circulation they are sufficiently accurate because every radiator has practically the same number of valves and tees in its circuit and therefore is affected equally by any variation from the ratios given above.

One elbow equivalent is approximately equal to the resistance offered by 25 diameters of straight pipe of the same nominal size. Hence, the equivalent length of one 2-in. diameter 90-deg. elbow is (25 × 2)/12 equals 4.2 ft. of straight pipe.

The resistance of a tee varies considerably with the relative proportions of the total flow entering or leaving the branch connection.[1] The values given in Table 53 for tees are for the pressure losses in the branch connections, and the pipe size of the equivalent elbows should be assumed to be that of the branch pipe. There is some loss of head in the straight-through flow, but it can safely be neglected.

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[1] See GIESECKE and BADGETT, "Supplementary Friction Heads in One-inch Cast Iron Tees," *Trans. A.S.H. & V.E.*, 1932.

Figures 159, 160, 161, and 162 give the friction heads for various pipe sizes and velocities. Note that the abscissas of all these charts are the heat conveyed per hour, based on a 20°F. temperature drop in the radiator. Figures 159 and 160 are based on the higher velocities used in forced-circulation systems, and Figs. 161 and 162 are for use in the calculation of gravity-circulation systems.

179. The Design of an Elementary Gravity-circulation System.—Consider the system shown in Fig. 149, in which the radiator is located directly above the boiler. In this case, the force maintaining circulation is the difference in weight between the two 10-ft. columns of 200° and 180° water. To avoid the labor of pressure-head calculations, Fig. 158 may be used thus: Find 200° in the upper margin, go vertically to the 180° line, and read, at the left margin, 90 milinches per ft. of water column. Multiplying this by the 10-ft. length of water column gives 900, the available pressure head.

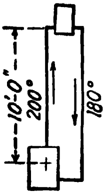

FIG. 149.—An elementary system.

After the available pressure head is found, the next step is to find what pipe size will produce that rate of circulation which produces a friction head of 900 milinches.

Assume that the actual length of pipe in the circuit is 26 ft. The number of elbow equivalents is calculated from Table 53 as follows:

	Elbow Equivalents
3 90-deg. elbows	3
1 radiator	3
1 boiler	3
Total	9

First try a $1\frac{1}{4}$-in. pipe. From Fig. 161 it will be found that a $1\frac{1}{4}$-in. pipe conveying 8,000 B.t.u. per hr. will give a friction loss of 2 milinches per ft. of pipe and 5 milinches per elbow. The total friction in the circuit would therefore be $(26 \times 2) + (9 \times 5) = 97$ milinches. Since the available head is 900 milinches, the $1\frac{1}{4}$-in. pipe is larger than necessary.

Next try a 1-in. pipe. The friction loss per foot of pipe and per elbow are, respectively, 8½ and 15½ milinches, and the total friction loss is 360.5 milinches, indicating that the 1-in. pipe also is larger than necessary.

A ¾-in. pipe gives a friction loss per foot of pipe and per elbow of 27 and 42 milinches, respectively, and the total friction loss is 1,080 milinches. The ¾-in. pipe is thus slightly small, and the 1-in. pipe would be used. If desired, part of the circuit could be made 1 in. and part ¾ in., and by several calculations the friction head could be made exactly equal to the available pressure head of 900 milinches, but such refinement is unnecessary.

180. Effect of Inexact Design.—The effect of an improper selection of pipe sizes will be examined. There are three possible ways in which errors can be made:

1. By making all parts of the system too small but of the proper relative size.
2. By making all the pipes too large.
3. By making the resistance of some circuits much greater than that of others.

If the pipe sizes are all *too small*, the effect will be to decrease the rate of circulation, and the radiators may receive insufficient heat. But the restriction of flow causes an increased temperature drop in the radiators, which increases the available circulating head and *tends* to maintain the velocity of flow. Unless the error is extreme, the system will therefore approach the performance set for it but with some decrease in the heat output of the radiators, because of their lower average temperature.

If the pipes are *too large* throughout, the primary effect will be to increase the rate of flow. This will cause a decrease in the temperature drop through the radiators, a reduction in the circulating head, and a consequent reduction of flow.

If a circuit to one radiator or group of radiators is undersized, the reduction in flow causes an increase in temperature drop, and the net result may be only a slight decrease in the heat delivery to the room.

It is thus apparent that gravity hot-water systems are to some extent *self-regulating*. It is because of this property that the ordinary hot-water systems, installed without exact design, operate satisfactorily. Indeed, for the usual small system it is not practicable to make exact calculations of the pipe sizes, experience having evolved empirical rules that give pipe sizes which are on the safe side and produce entirely acceptable results. While the heat delivered to the rooms may vary somewhat from the theoretical requirements, the error is well within that due to inaccuracies in computing the heat losses from the room.

In large installations, the exact method has some distinct advantages. The liberality with which the pipe sizes of a small system are selected cannot be practiced on a large system without a considerable increase in the cost of the installation, whereas any pipes that may be chosen too small can be replaced only at great expense. Throttling valves, although they should be placed on the branch circuits as a precaution, are difficult to adjust and are easily tampered with. Hence a calculation of the pipe sizes in the manner outlined is desirable for large or important installations.

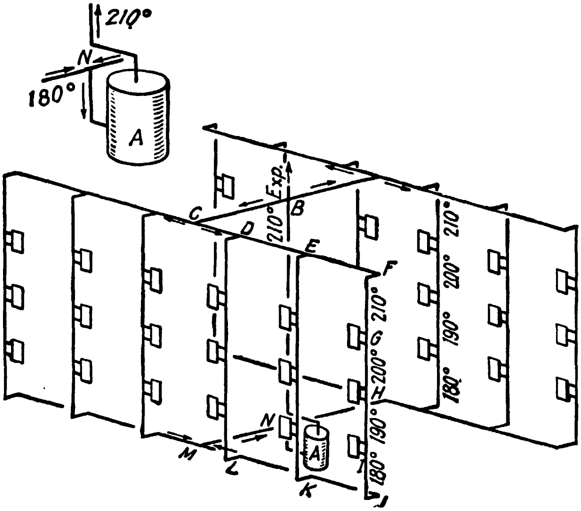

FIG. 150.—A gravity-circulation system for a three-story building.

181. Design of a Heating System for a Three-story Building.—The system is to have a gravity circulation with overhead distribution, as shown in Fig. 150.

To simplify the design, there are to be 12 radiators on each floor, each to dissipate 8000 B.t.u. per hr. The boiler or heater is to be in the basement, the supply riser at the center of the building, and the supply and return mains are to branch as shown, so as to divide the heating system into four equal parts.

The flow lines are to be well covered, so that the cooling in them may be neglected.

The pressure head maintaining circulation through any one of the 12 return risers is produced by the difference in the weight of the water in the supply riser and that in the respective return risers.

TABLE 54

Section	B.t.u. in 1,000	Feet, elbows	Pipe size, in.	Unit friction	Total friction
AB	. . .	44	4	4	176
	288	2	4	29	58
BC	. . .	20	2½	12	240
	144	4	2½	57	228
CD	. . .	8	2½	4	32
	72	4	2½	15	60
DE	. . .	15	2	4	60
	48				
EF	. . .	15	1¼	8	120
	24				
FG	. . .	13	1¼	8	104
	24	2	1¼	19	38
GH	. . .	9	1¼	8	72
	24				
HI	. . .	9	1¼	8	72
	24				
IJ	. . .	5	1¼	8	40
	24	2	1¼	19	38
JK	. . .	15	1¼	8	120
	24				
KL	. . .	15	2	4	60
	48				
LM	. . .	8	2½	4	32
	72				
MN	. . .	20	3	4	80
	144	4	3	23	92
NA	. . .	8	4	4	32
	288	8	4	29	232
Total. . . .	. . .	204			
	. . .	26	. . .	. . .	1986

NOTE.—Available pressure head, 2,136 milinches.

Assume the temperature drop to be 30°, the water leaving the boiler or heater at 210° and returning at 180° when every radiator is dissipating 8000 B.t.u.

Under these conditions the water will reach the third-floor radiator at a temperature of 210°; after the water returning from the third-floor radiator has mixed with the water in the return riser, the resulting temperature in the riser will be 200°, and the water will reach the second-floor riser at that temperature; similarly, the water will reach the first-floor radiator at a temperature of 190° and the boiler or heater at a temperature of 180°, if the cooling in the return main is neglected.

The pressure head will be produced by a column of water in the supply riser 24 ft. high at a temperature of 210° and a corresponding column of water in the return riser, of which 9 ft. is at a temperature of 200°, 9 ft. at 190°, and 6 ft. at 180°. From Fig. 158, the unit pressure heads for these cases are 50, 94, and 140 milinches. The total pressure head is

$$(9 \times 50) + (9 \times 94) + (6 \times 140) = 2{,}136 \text{ milinches.}$$

Next record the available pressure head in Table 54 and the designations and descriptions of the 14 sections into which the system is divided. Begin the design with that riser which is located most unfavorably, that is, which will be most difficult to supply with its proper portion of hot water—in this case the one farthest from the supply riser.

There are about 204 ft. of pipe and about 26 elbow equivalents in the circuit of the far riser. Placing **one elbow equivalent to 4 ft. of pipe,**[1] the circuit is equivalent to 308 ft. of pipe. If the pressure head of 2,136 milinches is distributed uniformly along the circuit, the several pipe sizes must be such that the *average* friction head is 2,136 divided by 308, or 7.0 milinches per ft.

Select the pipe sizes from Figs. 161 and 162 as follows: Section *AB* conveys 288,000 B.t.u. with a temperature drop of 30°. Since Figs. 161 and 162 were constructed for a temperature drop of 20°, locate two-thirds of 288,000, or 192,000 B.t.u. in Fig. 162 and note that for a friction head of 4 milinches a 4-in. pipe should be used. Record this size in the table, select the remaining 13 sizes in a similar manner, record them as shown, and find that the total friction head will be 1,986 milinches. This is about 7 per cent less than the available head and is therefore safe. If section *NA* were 3½-in. pipe, the agreement would be closer, but too much refinement is useless, and furthermore 3½-in. pipe is seldom used in heating work.

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[1] See footnote, bottom of Table 53.

To design the middle return riser *EK*, note that it conveys 24,000 B.t.u., is composed of 36 ft. of pipe with about 10 elbows, and must have the same friction head as the circuit *EFJK*, or 604 milinches. The circuit must evidently be composed partly of 1¼-in. and partly of 1-in. pipe.

The third return riser *DL* is designed in a similar way, so as to have the same friction head as the circuit *DFJL*, or 724 milinches.

To design the radiators and their connections to the return risers, note that there are two pressure heads tending to cause the water to flow through the radiators. One is the difference in pressure heads in the return riser between the two points at which the radiator is connected to the riser; it is 8 milinches per ft. according to Table 54. The second is that produced by the difference in weight of the column of hot water in the riser and that of the cooler water in the radiator.

Select a 38-in. radiator and space the riser connections 34 in. on centers. Select 1¼-in. pipe for the radiator connections, which will consist of 2 ft. of pipe. For a top-floor radiator the other resistances are

	Elbows
1 tee, 40 per cent through branch	8
1 radiator valve	12
1 radiator	3
1 tee, 40 per cent through branch	8
Total	31

For a temperature drop through the radiator of 20° the friction head will be 159 milinches;[1] and the pressure head, 164 milinches.[2] This is satisfactory. The radiator size would be calculated for an assumed temperature drop of 20° in the radiator.

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[1] From Fig. 161, unit friction head 1¼-in. pipe = 2 milinches and elbow 5 milinches; (2 × 2) + (5 × 31) = 159.

[2] *I.e.,* 50 milinches per ft. in radiator (from Fig. 158, supply 210° and return 200°)
8 milinches per ft. in riser (from Table 54, *FG* equals 8 milinches)
58 milinches per ft. total
$58 \times \frac{34}{12}$ ft. = 164 milinches total

182. Summary.—The steps taken in selecting the pipe sizes may be stated as follows:

1. Having calculated the required B.t.u. output of the radiators, make sketch of piping layout and designate by numbers or letters the sections of the main supply riser and the longest route that water must follow.
2. Select temperature drop and compute temperature of water at various points.
3. Calculate total pressure head, which is the head available to produce circulation.
4. Select pipe sizes, for the longest route, that will give an over-all friction drop nearly equal to the available pressure head.
5. Select pipe sizes for the remaining risers so that the friction drop through all parallel circuits will be nearly equal to the respective available pressure heads across them.
6. Select pipe sizes for radiator connections.

183. Forced-circulation Systems.—The advantages of using a pump to produce the circulation are that much smaller pipes can be used and a positive circulation is assured throughout the system. In large installations, especially where more than one building is served, forced circulation is almost invariably adopted; and with the development of satisfactory circulating pumps of moderate cost it is being used more and more in residence heating instead of gravity circulation.

The circulating head produced by a pump is so much greater than the gravity head that the latter can usually be neglected in the calculations. Instead of an available head of, say, 2 or 3 in. water, a forced-circulation system using centrifugal pumps may have a head of around 20 lb., which is more than 200 times the gravity head. There are, however, certain types of circulator which are intended merely to remedy a poorly designed system and which develop only small pressure heads, although these are still considerably greater than the gravity head.

There are some essential differences in the circulation in a forced-circulation system as compared with the gravity system. It is necessary to understand what takes place in order to design the system properly and to remedy cases of trouble that are encountered in the field. The most important point is that in the forced-circulation system the circulating force (neglecting the gravity effect) is produced at a single point, namely, at the pump, whereas in the gravity system the circulating force is produced uniformly throughout the vertical portions of the piping. Another difference is that the self-regulating property of the gravity system is almost lacking in the forced-circulation system, so that if the pipe sizes in the latter are greatly restricted at some points the effect is likely to be serious.

The general principle of design in the forced-circulation system is somewhat similar to that in the gravity system. The circulating head is first selected which varies from 2 to 5 ft. on small-size systems and increases for larger buildings. Next, the available pressure drop per foot of pipe is established which should not exceed from 240 to 360 milinches for the average system, and for large industrial buildings it may range from 480 to 720 milinches per foot. The pipe sizes are then chosen so as to give a pressure drop that will be approximately equal to the available circulating head. The pump capacity may be determined from Eq. (7).

$$Q = \frac{H}{(t_f - t_r) \times 60 \times 8} \tag{7}$$

Where Q = volume water circulated, gal. per min.
H = heat delivered, B.t.u. per hr.
t_f = flow water temperature, deg. F.
t_r = return water temperature, deg. F.
60 = minutes in 1 hr.
8 = weight 1 gal. water at 215°.

From Eq. (7) it will be observed that if a 20° drop in temperature is used, $20 \times 60 \times 8 = 9{,}600$. Hence, for quick estimating purposes it may be assumed that 1 gal. per min. of water will be required for each 10,000 B.t.u. of capacity.

Example.—Assume the length of a forced hot-water reversed return system (Fig. 147*b*) is 150 ft. and the equivalent length of boiler and fittings is 100 ft. The total radiation load is 98,000 B.t.u. per hr. Calculate the pump capacity and head required if the flow-water temperature is 210° and the return 190°. Also determine the size of supply pipe leading from the boiler..

Substituting in Eq. (7)

$$Q = \frac{98{,}000}{(210 - 190) \times 60 \times 8} = 10.2 \text{ gal. per min.}$$

Therefore the pump must be capable of delivering 10.2 gal. per min. for the (150 + 100) 250 ft. of equivalent length of pipe. For this size of system it is customary to allow for a friction head of 240 milinches per ft. The total friction head would be 250 × 240, or 60,000 milinches, which is equal to 60,000/12,000 or a 5-ft. head, for which the pump should be selected. From Fig 159 it will be found that a 1¼-in. pipe will be required to convey 98,000 B.t.u. per hr. at a friction head of 240 milinches per ft.

In gravity hot-water systems it is customary to select the radiation sizes on a heat-emission rate of 150 B.t.u. per hr. per sq. ft., which is based on a flow or supply water temperature of 180° and a return temperature of 160°. However, in a forced circulation system using a closed expansion tank (Art. 187), it is possible to vary the water temperature over a wide range. Data in Table 55 give heat-emission rates for various average radiator temperatures, from which the radiation capacity may be readily computed after the heat loss or capacity is determined in B.t.u. per hour.

TABLE 55.—HEAT-EMISSION RATES FOR CAST-IRON RADIATION

Mean Radiator Temp., deg. F.	Heat Output, B.t.u. per Sq. Ft. Radiation
160	130
170	150
180	170
190	190
200	210
210	230
215	240
220	250

184. Small Forced-circulation System.—In recent years copper tubing has come into rather common use for hot-water heating systems where the pipe sizes are 2 in. or less. Concurrently a number of small circulating pumps have been developed, and the one-pipe system of Fig. 147*c* has been studied and improved. As a result a very satisfactory system, particularly for residential work, is being advocated by various manufacturers. A circulating pump of the form commonly used is shown in Fig. 151. Special tees are sometimes used, which assist flow to the radiators and tend to cause the cooler water from each radiator to flow along the bottom of the supply main. A system of this sort is illustrated in Fig. 152.

For small residences having a heat loss of less than 60,000 B.t.u. per hr., a design procedure has been developed using pipe and fittings of the standard *screwed* type. The circuit layout is comparable to the arrangement shown in Fig. 152, with the exception that the trunk supply main is 1 in. and each circuit is sized at ¾ in. Diagrammed in Fig. 153 are the radiator connec-

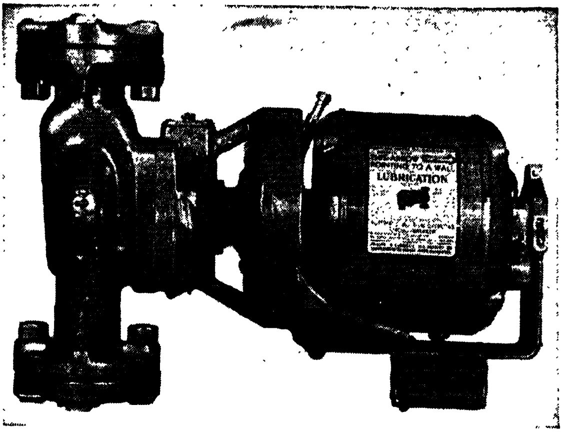


FIG. 151.—Small circulating pump. (*Bell and Gossett Company.*)

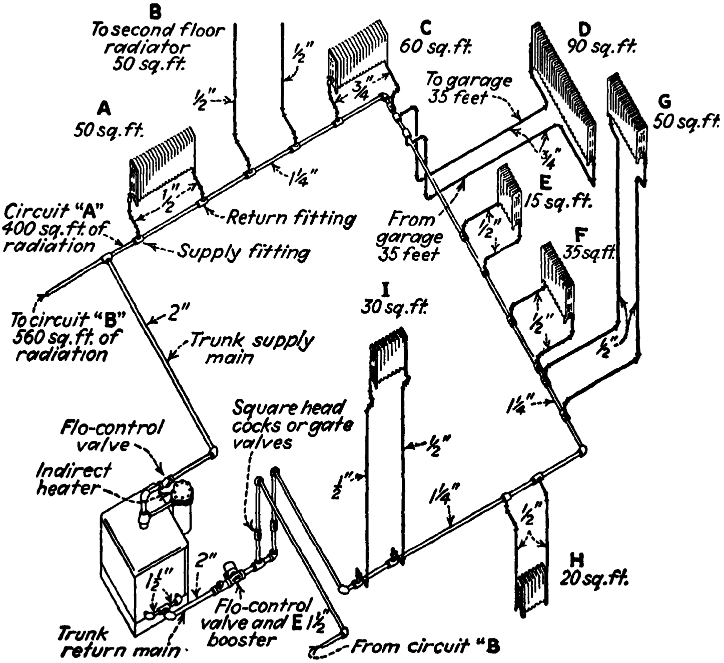
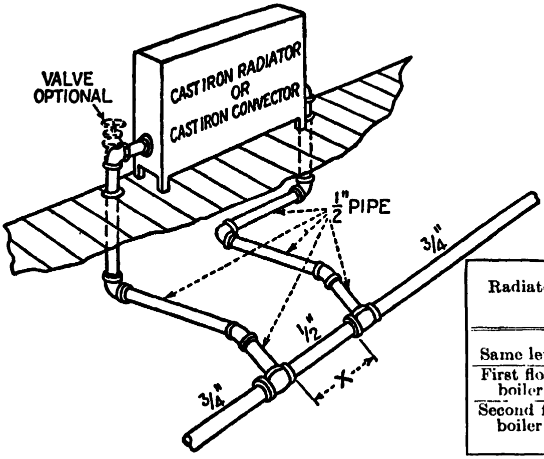


FIG. 152.—Small single-main forced-circulation system using special radiator-connection tees.



Radiator location	Distance X, ft.
Same level as boiler	2.5
First floor above boiler	1.0
Second floor above boiler	2.5

FIG. 153.—Radiator connections for one-pipe forced-circulation system using standard screw fittings.

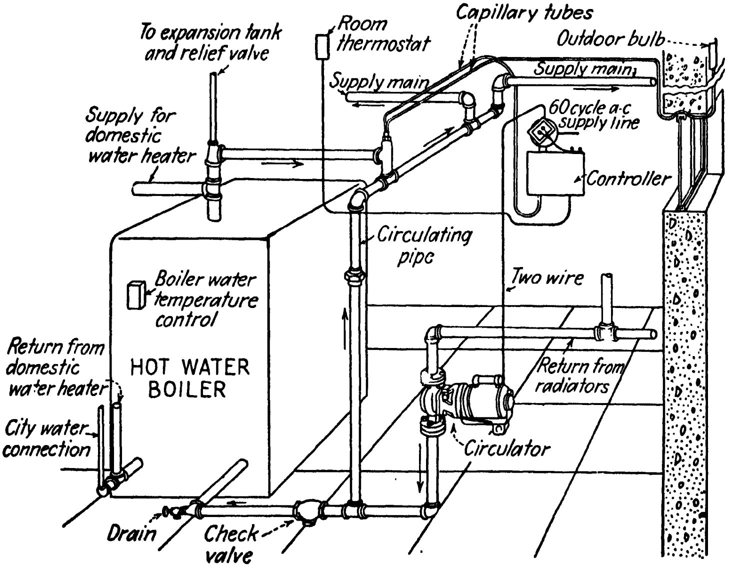


FIG. 154.—Control with boiler by-pass arranged for forced circulation.

tions, which show that section *X* between the two branches is variable depending upon the location of the radiator.

In such systems temperature control in the house may be simply accomplished by having the thermostat start and stop the circulating pump, and a device such as a weighted flapper valve stops circulation when the pump is not running. A valve of this type also prevents circulation of hot water to the radiators during summer operation of an indirect domestic hot-water heater.

Starting and stopping the pump has certain inherent disadvantages in that the room temperature tends to fluctuate during the on-and-off cycles. This condition may be partially overcome by using a control arrangement similar to that shown in Fig. 154, which permits to the system countinuous circulation of water at varying temperatures. This control consists of a three-way valve, which is designed to by-pass a varying portion of the returned water to the heating system and automatically add the proper amount of heated water from the boiler to balance the heat loss of the system.

A liquid expansion bulb mounted on an outside wall, as indicated in Fig. 154, hydraulically adjusts the opening of the three-way valve to compensate for outdoor temperature fluctuations. As there is a certain lag between changes in atmospheric conditions and indoor temperatures, it is also advisable to use an electric room thermostat to prevent any overheating. This thermostat functions, by means of a heater coil located in the controller, to cause another liquid expansion bulb also to act on the three-way valve in such a way as to prevent further hot water from being drawn from the boiler.

This control arrangement effectively checks overheating, at the same time maintaining the responsiveness of the system to outside temperature changes, because circulation is not interrupted. The boiler water is maintained at a constant temperature at all times by the firing device.

There are various possible ways of accomplishing mechanically (or electrically) this general method of control, and the device described is only one of them. The objectives of this method are (1) control of the water temperature in accordance with the outdoor temperature (which is a rough, over-all control) and (2) the superimposing upon that rough control of a direct control responsive to the room temperature.

Figures 163 and 164 (at end of this chapter) give the friction loss in copper tubing and fittings for different velocities of flow. The curves in Fig. 164 have been extrapolated below a velocity of 20 in. per sec. These two charts are based on velocity of flow rather than heat conveyed per hour as are the charts for iron pipes. In using them it is therefore necessary to calculate the velocity of flow as follows:

Let

v = velocity of flow, in. per sec.
d = internal pipe diameter, in.
D = density of water, lb. per cu. ft.
H = heat delivered per hr., B.t.u.
t_f, t_r = flow and return temperatures, deg. F.
W = weight of water flowing, lb. per hr.

Then

$$H = W(t_f - t_r)$$

$$W = \frac{V}{12} \times 3{,}600 \times \frac{\pi d^2}{4} \times \frac{1}{144} \times D$$

$$W = 1.635vd^2D \tag{8}$$

Assuming an average value of $D = 60.68$,

$$W = 99.21vd^2 = \frac{H}{t_f - t_r} \tag{9}$$

$$v = 0.01008 \frac{H}{(t_f - t_r)d^2} \tag{10}$$

In cases where $t_f - t_r = 20°$, the formula becomes

$$v = 0.000504 \frac{H}{d^2} \tag{11}$$

185. Pipe-size Tables.—Although the method of correctly designing hot-water heating systems is easily applied, tables of pipe sizes may be useful at times. For that reason, Tables 56 and 57 were prepared to give the pipe sizes for one-pipe gravity-circulation heating systems like that described in Art. 176 and suitable for residences and similar buildings of ordinary size.

Designers must be careful to use these tables only for heating systems of the kind and size for which the tables were prepared. All heating systems of unusual type or size should be designed by means of the more thorough method.

TABLE 56.—MAXIMUM CAPACITIES, IN 1000 B.T.U., OF MAINS FOR ONE-PIPE SYSTEMS FOR A DROP IN TEMPERATURE OF 25° AND A HEAD[1] OF 3 FT. 6 IN.

Lengths of horizontal mains, ft.	2 in.	2½ in.	3 in.	3½ in.	4 in.	5 in.	6 in.
50	41	59	100	140	195	346	552
75	35	51	90	130	179	320	500
100	31	48	83	121	165	293	450
125	30	44	75	113	153	280	425
150	27	40	70	104	143	265	405
175	25	38	66	98	136	252	390
200	24	35	63	93	130	239	377
225	23	34	61	90	125	225	367
250	22	33	60	88	120	223	358

NOTE.—For given capacities, the permissible lengths of the mains are practically directly proportional to the heads. If the head is more than 3 ft. 6 in., the values given in this table may be increased correspondingly.

[1] From mains to center of boiler.

TABLE 57.—MAXIMUM CAPACITIES, IN B.T.U., OF RISERS FOR ONE-PIPE SYSTEMS

Flow riser, in.	Return riser, in.	Capacities in B.t.u.		
		1st floor	2d floor	3d floor
¾	¾	3,000	6,500	7,880
¾	1	3,700	8,000	9,600
1	1	5,300	11,400	13,600
1	1¼	6,400	14,400	17,100
1¼	1¼	9,500	21,700	26,300
1¼	1½	10,700	25,000	30,600
1½	1½	13,000	30,200	37,900
1½	2	16,000	38,100	46,100
2	2	23,000	55,700	66,800

NOTE.—Length of pipe, first floor, 7 ft. 0 in.; second floor, 25 ft. 0 in.; third floor, 43 ft. 0 in. Each circuit has 16 elbow equivalents.

This table is based on a temperature drop of 20° through the radiators and an average water temperature of 180°. The resulting pressure heads are as follows: first floor, 129 milinches; second floor, 903 milinches, third floor, 1,677 milinches.

186. Expansion Tanks.—Provision must be made in every heating system for the changes in the volume of the water to take place without injury to the system or to its operation. This is generally done by means of an open expansion tank, as shown in Fig. 155.

To find the proper size of the expansion tank, determine the volume of water necessary to fill the entire system, that is, the boiler, the radiators, and all pipe lines; divide this volume by 25 (because 25 cu. ft. of water at 40° will expand to about 26 cu. ft. at 200°); the result represents the volume increase that needs to be provided. The actual size of the expansion tank should be about double this figure.

An approximate rule for the size of the expansion tank is that the number of square feet of radiation in the system divided by 40 equals the capacity of the expansion tank in gallons.

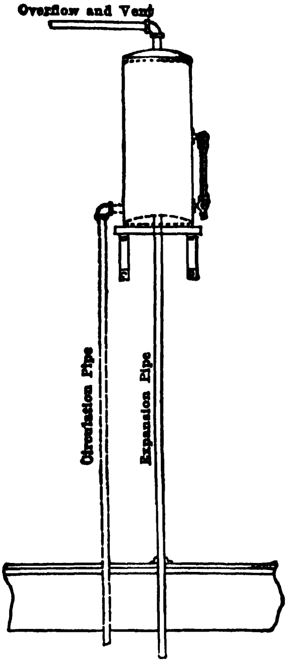


Fig. 155.—Arrangement of expansion tank—open system.

In forced-circulation systems, the open expansion tank should be connected to the suction side of the pump and as near to it as practicable.

In gravity-circulation systems, the expansion tank should be connected to the flow riser so that air bubbles liberated from the hot water in the heater may escape readily through the expansion tank.

In every case the expansion tank should be placed at such an elevation above the highest radiator that the water in that radiator will be under sufficient pressure to prevent boiling at the maximum temperature at which the water is to be used. The expansion tank should be located and protected so that the water in the tank or in the connecting pipe lines cannot freeze. If the water should freeze in the expansion tank or in the connecting pipe lines, and the water in the system heats and causes further expansion, the resulting force will burst the boiler or some of the radiators.

187. Closed Systems.—The ordinary expansion tank is open to the atmosphere and located above the highest radiator. If it is replaced by a closed tank located at a lower elevation and partly filled with water and partly with air so that, as the volume of water in the system expands, the volume of air in the closed tank will be compressed without producing undue pressures in the system, the resulting system is called a "closed system." The ordinary system is called an "open system." In some closed systems the pneumatic tank is supplemented by a pressure-reducing valve and a pressure-relief valve which will prevent excessive internal pressures in the system (Fig. 156).

The system shown in Fig. 156 is frequently used in residences. It is fully automatic, since the pressure-reducing valve admits water and the relief valve discharges water so as to maintain a constant pressure. The tank conserves water but may be omitted.

Fundamentally, the open and closed systems are exactly alike. As ordinarily installed, a closed system provides higher water pressures than an open system and consequently permits higher water temperatures without danger of boiling. The higher water temperatures, in turn, permit higher temperature drops through the radiators and, consequently, smaller pipe sizes and smaller radiators. For example, with sufficient pressure in the system, water could be sent to the radiators at 240° and returned at 210°; the average water temperature in the radiator would be 225°, and the radiators would be smaller than corresponding steam radiators. From a comfort standpoint, however, high radiator temperatures are undesirable in a hot-water heating system.

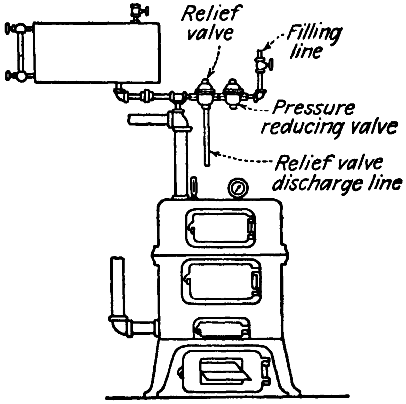


FIG. 156.—Arrangement of closed system.

188. Power for Pumping.—A simple calculation gives the power required for pumping.

$$Hp = \frac{PV}{33{,}000 \times e} \tag{12}$$

where Hp = horsepower at the pump shaft.
P = pressure head, lb. per sq. ft.
V = volume of water pumped, cu. ft. per min.
e = efficiency of pump.

The efficiency of the pump will ordinarily be between 60 and 70 per cent.

Example.—Calculate the horsepower required for the pump in a system having a friction head of 3.99 lb. per sq. in., a B.t.u. load of 288,000, and a temperature drop of 30°.

$$P = 3.99 \times 144 = 574.56 \text{ lb. per sq. ft.}$$

The *weight* of water circulated is $\frac{288,000 \text{ (B.t.u.)}}{(30 \text{ (deg.)}} = 9,600$ lb. per hr.

$$V = \frac{9,600}{60 \times 60.58} = 2.64 \text{ cu. ft. per min.}$$

It is assumed that the pump is to be placed in the main return pipe, and the water density is therefore taken as that corresponding to 180°, the return temperature.

$$Hp = \frac{574.56 \times 2.64}{33,000 \times 0.65} = 0.0707$$

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Problems

1. For the system shown in Fig. 149 assume that the vertical dimension is 6 ft. (instead of 10 ft.) and that the radiator emits 12,000 B.t.u. per hr. with a temperature drop from 190 to 170°. What pipe size should be used?

2. If 1-in. pipe were used for the system shown in Fig. 149, how much heat would the radiator dissipate with temperatures of 200 and 180° in the risers? Find this by trial from the diagram of Fig. 161.

3. If 1¼-in. pipe were used and the radiators moved horizontally so as to increase the radiator circuit to 46 ft. of pipe and 13 elbow equivalents, how much heat would the radiator dissipate?

4. For the system shown in Fig. 150, operating as a gravity system, select the pipe sizes for the circuit *DL*, using the temperatures and radiator heat dissipation as given in the text.

5. For the system shown in Fig. 150, operating as a forced-circulation system, select the pipe sizes for the circuit *ABCFJMNA*, using a friction drop of 1 lb. per sq. in. and with the temperatures and radiator heat dissipation as given in the text.

6. What horsepower would be required for the pump in a heating system delivering 1,450,000 B.t.u. per hr. with a 20° temperature drop, a pressure drop in the system of 10 lb. per sq. in., and the pump having an efficiency of 70 per cent, located in the return main (temperature 170°)?

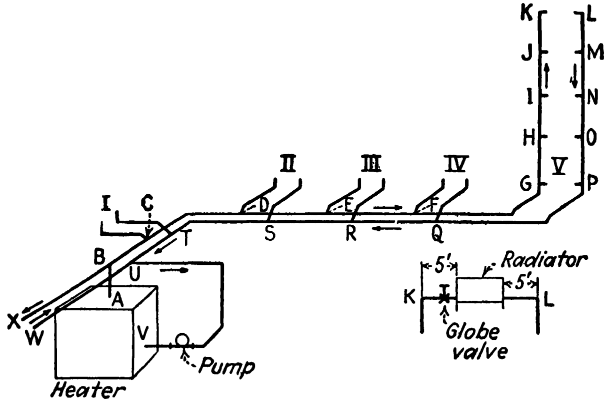

FIG. 157.

7. Calculate the pipe sizes for the circuit *ABGKLPUV* in Fig. 157. This is a forced-circulation system, using steel pipe, and the pump is designed for a head of 5 lb. The temperature drop in the radiators is to be 20°. The heat conveyed through each section is as follows:

	B.t.u. per Hr.
BX and *WU*	510,000
Riser I	40,000
Riser II	80,000
Riser III	80,000
Riser IV	75,000
Radiator G	13,000
Radiator H	13,000
Radiator I	13,000
Radiator J	13,000
Radiator K	16,000

The lengths of pipe in the various sections are

	Ft.		Ft.		Ft.		Ft.
A-B	5	*G-H*	13	*M-N*	13	*S-T*	23
B-C	15	*H-I*	13	*N-O*	13	*T-U*	12
C-D	22	*I-J*	13	*O-P*	13	*U-V*	25
D-E	20	*J-K*	13	*P-Q*	32		
E-F	20	*K-L*	10	*Q-R*	20		
F-G	31	*L-M*	13	*R-S*	20		

TABLE 58.—DENSITY OF WATER AT VARIOUS TEMPERATURES[1]
(According to Thiesen, Scheel, Dieselhorst, Hirn, Ramsay and Young, and others)

Temperature, °F.	Density, lb. per cu. ft.	Temperature, °F.	Density, lb. per cu. ft.	Temperature, °F	Density, lb. per cu. ft.
32	62.42	92	62.10	152	61.16
34	62.42	94	62.06	154	61.12
36	62.42	96	62.02	156	61.08
38	62.43	98	62.03	158	61.04
40	62.43	100	62.00	160	61.00
42	62.43	102	61.98	162	60.96
44	62.42	104	61.95	164	60.92
46	62.42	106	61.92	166	60.88
48	62.41	108	62.89	168	60.84
50	62.41	110	61.86	170	60.80
52	62.40	112	61.83	172	60.76
54	62.39	114	61.80	174	60.71
56	62.39	116	61.77	176	60.67
58	62.38	118	61.74	178	60.62
60	62.37	120	61.71	180	60.58
62	62.36	122	61.68	182	60.54
64	62.35	124	61.65	184	60.49
66	62.33	126	61.62	186	60.45
68	62.32	128	61.59	188	60.40
70	62.30	130	61.55	190	60.36
72	62.29	132	61.52	192	60.31
74	62.27	134	61.49	194	60.26
76	62.26	136	61.45	196	60.22
78	62.24	138	61.42	198	60.17
80	62.22	140	61.38	200	60.12
82	62.20	142	61.35	202	60.07
84	62.18	144	61.31	204	60.02
86	62.16	146	61.28	206	59.98
88	62.14	148	61.24	208	59.93
90	62.12	150	61.20	210	59.88

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[1] Translated and developed from Hutte by F. E. Giesecke.

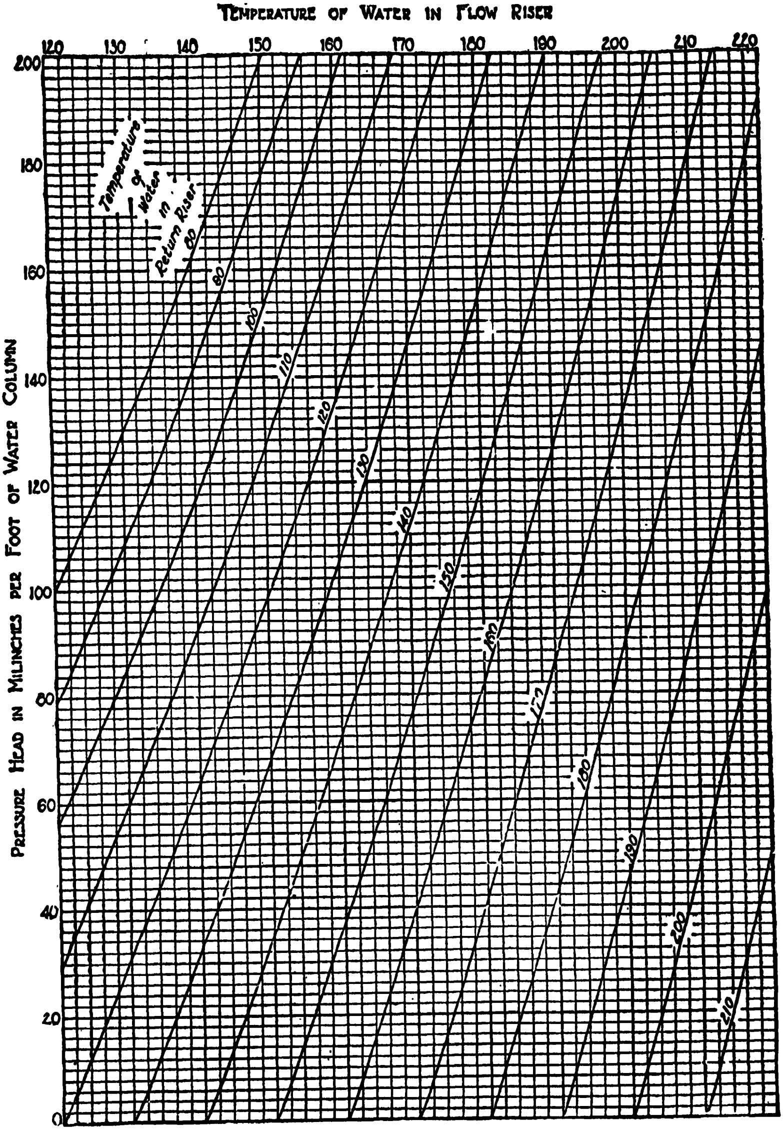

Fig. 158.—Pressure heads for various temperature differences.

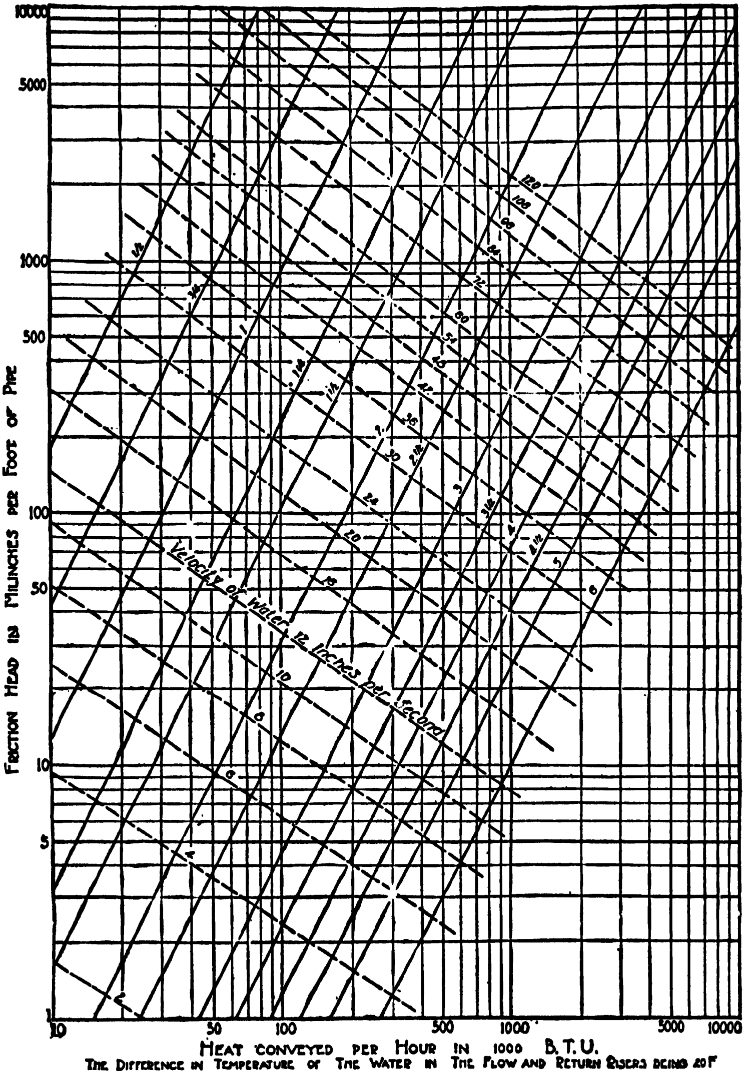

Fig. 159.—Friction heads in pipes for a temperature difference of 20°.

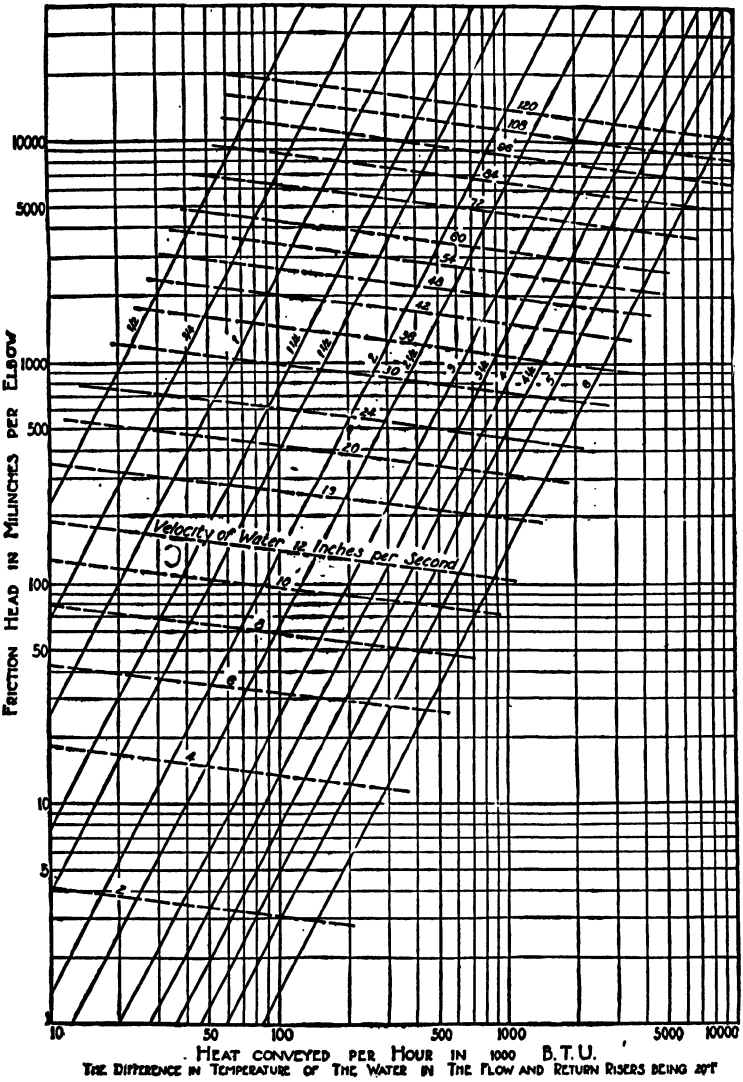

Fig 160.—Friction heads in elbows for a temperature difference of 20°.

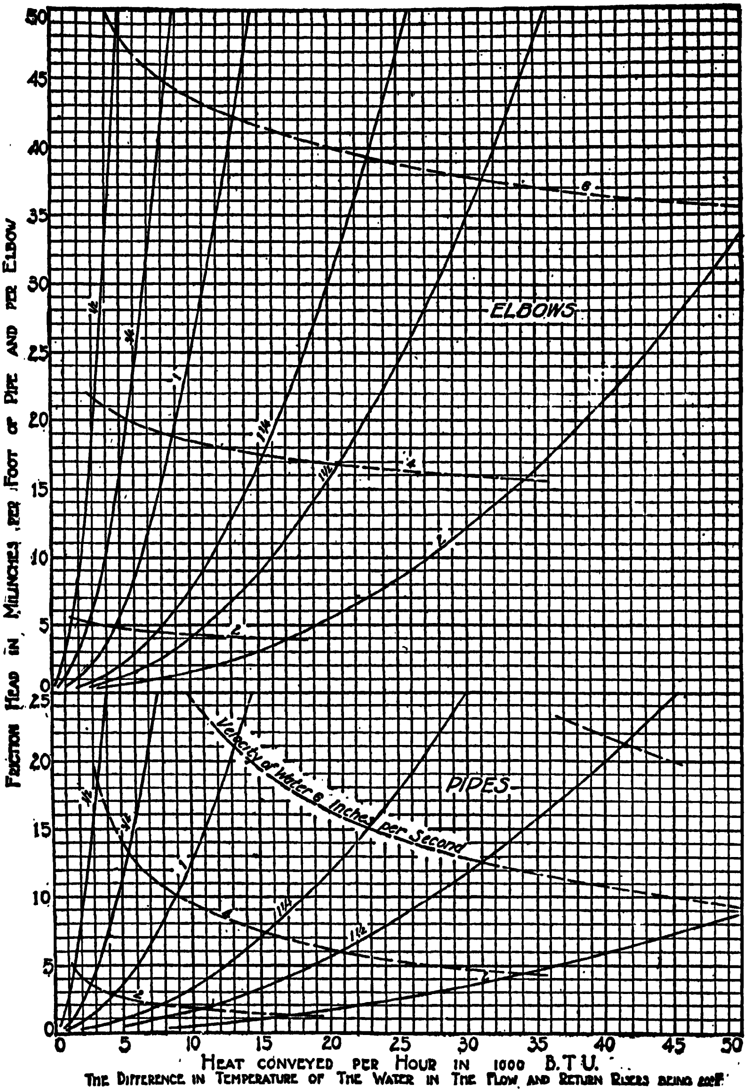

Fig. 161.—Friction heads in pipes and elbows for a temperature difference of 20°.

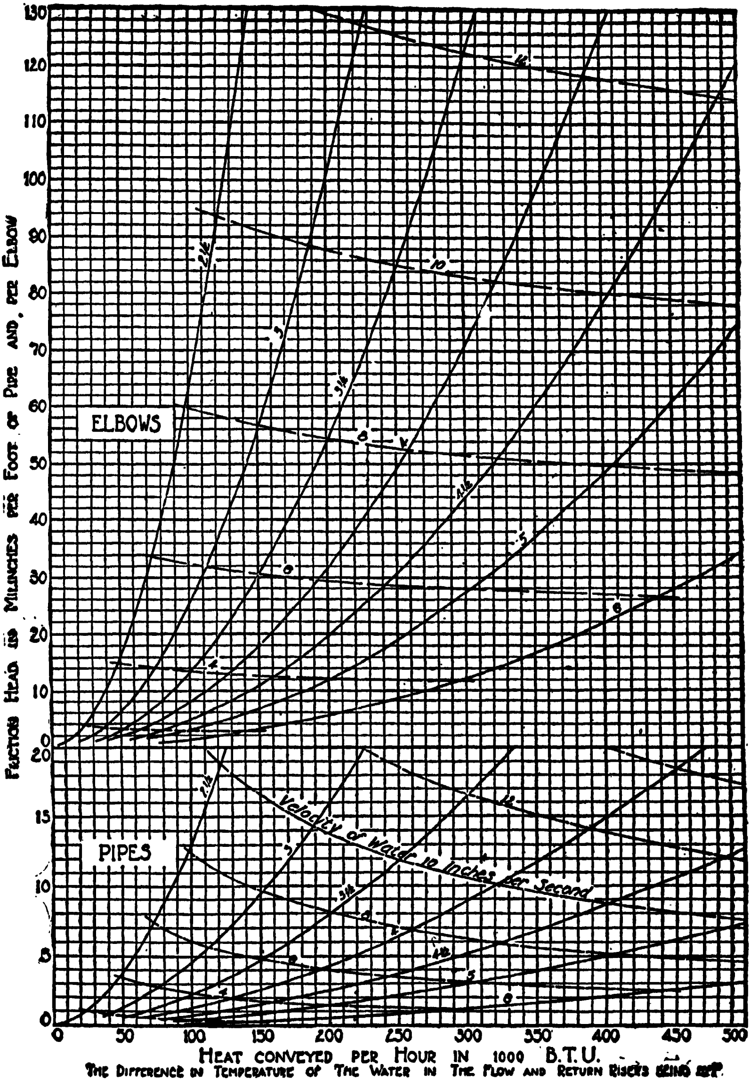

Fig. 162.—Friction heads in pipes and elbows for a temperature difference of 20°.

The problem is not set forth in extreme detail as to the exact pipe fittings but is as exact as most actual projects are when the pipe sizes are computed. Judgment should be used in allowing for fittings. First estimate the over-all equivalent length of the circuit, and find an approximate average friction loss per foot.

3. What is the friction loss in a circuit consisting of a boiler, one radiator, four copper elbows, and 39 ft. of 1-in. copper pipe? The temperature drop in the radiator is 30°, and the radiator is designed to dissipate 47,900 B.t.u. per hr.

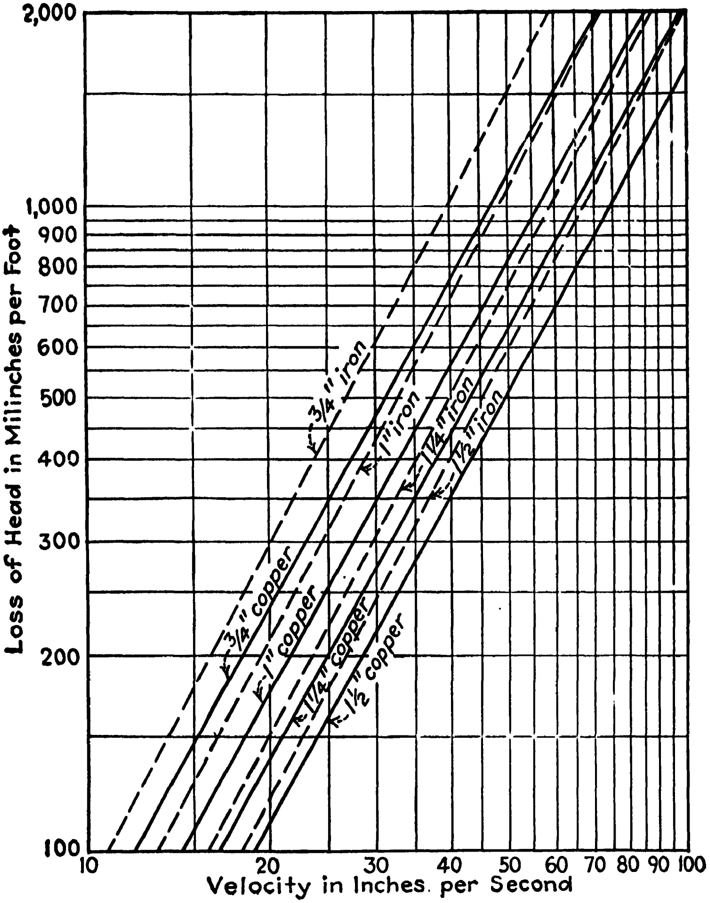

FIG. 163.—Loss of head in various sizes of iron and copper pipe. (The copper pipe has actual diameters corresponding to type *M* pipe. See Chap. VIII.) (*F. E. Giesecke and W. H. Badgett, Trans. A.S.H. & V.E.*, 1932.)

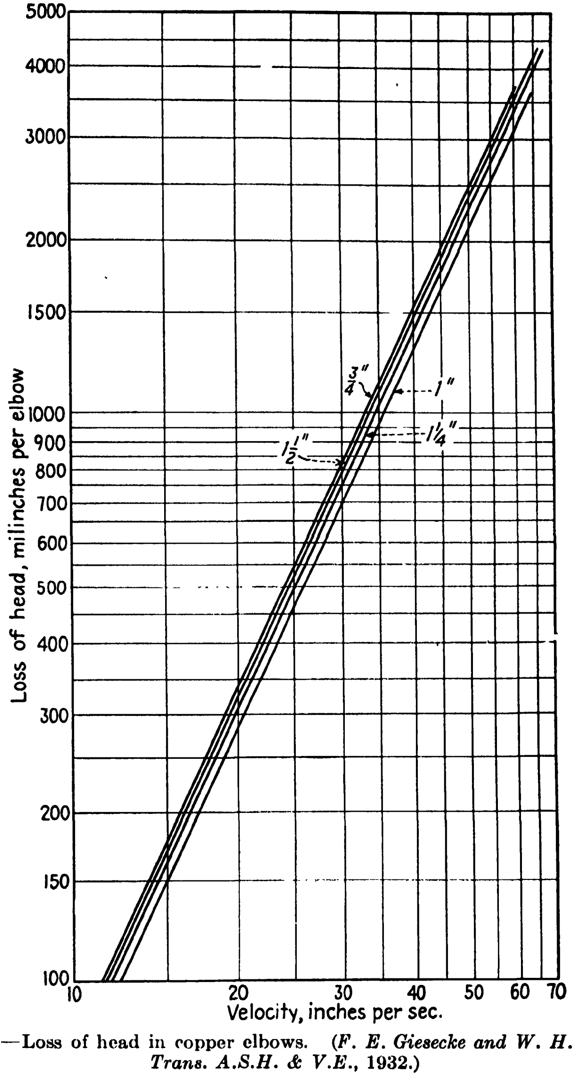

FIG. 164.—Loss of head in copper elbows. (*F. E. Giesecke and W. H. Badgett Trans. A.S.H. & V.E.*, 1932.)

CHAPTER XI

AUTOMATIC CONTROL

189. Importance of Automatic Control.—The discussion thus far has been concerned almost entirely with the design of heating systems from the standpoint of *adequacy* to meet the most severe conditions. Of no less importance, however, is the ability of a heating system to respond to the changing requirements imposed upon it by outside-temperature fluctuations. This feature of heating-system design has not been given the attention it needs, but at present much thought is being devoted to the improvement of control methods and devices. A temperature-control device, instead of being regarded merely as an accessory, is becoming recognized as an integral part of the heating system.

In air conditioning, control devices are becoming more and more important because requirements for more exact conditions are steadily being stiffened. In fact, the success or failure of many installations has been directly chargeable to the control apparatus.

In this chapter the fundamentals of control systems and certain applications will be discussed, but more specific applications are considered in later chapters.

190. The Elements of a Control System.—Control systems consist of one or more **primary elements,** which are sensitive to temperature, pressure, or humidity. From the action of these primary elements **secondary elements** such as valves, air dampers, or electric switches are caused to open, close, or change position. In general, the forces developed by the primary elements are insufficient to operate the valves, dampers, or switches, and it is therefore necessary to use an auxiliary motive power, compressed air, or electricity, for that purpose. The primary element merely operates a small compressed-air valve or interrupts or modifies an electric circuit, which in turn controls the action of the secondary elements.

Many ingenious arrangements have been devised for the combining of two or more primary elements. In many ways electrical systems offer greater possibilities for certain applications, and in recent years there has been great interest in the development and use of electrical controls. Compressed air, however, is quite satisfactory for many cases and has certain advantages in simplicity and ease of maintenance.

191. Primary Control Elements.—The most important of the primary elements is the temperature-control element, which has several forms. The bimetallic strip is a common type. It consists of two strips of metal, usually steel and brass, brazed together. These metals have widely different coefficients of expansion, and small changes of temperature cause the strip to change its shape.

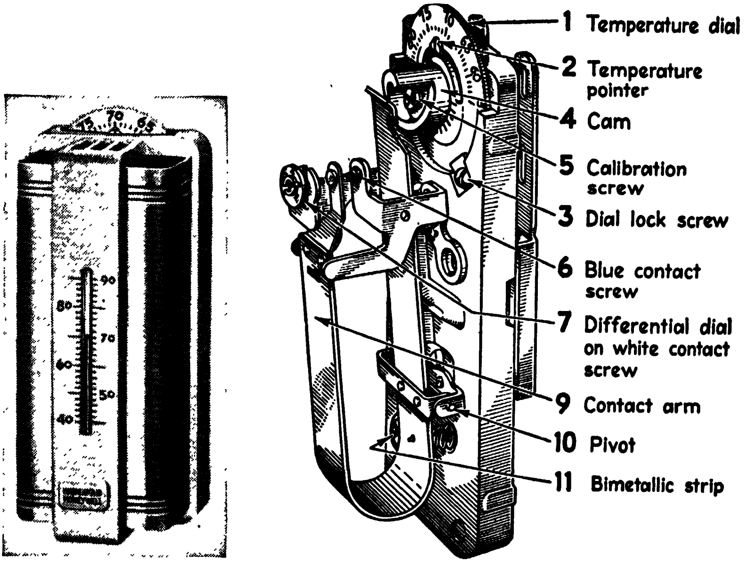

FIG. 165.—Electrical type, bimetal room thermostat (with and without cover). (*Minneapolis-Honeywell Regulator Company.*)

Other temperature elements are gas- or liquid-filled bulbs or materials similar to hard rubber, which have a high coefficient of thermal expansion. A common application of the bimetallic strip in the simple electric thermostat operating the dampers of a boiler is shown in Figs. 165 and 166.

The bimetallic strip 11 shown in Fig. 165 is pivoted at point 10, and contact arm 9 is deflected from front to back by the unequal expansion of the metals with change in temperature. The thermostat is connected electrically with a motor in such a way that, as the temperature drops and strip 11 makes contact with 7, a current of low voltage is transmitted through the circuit. By means of a relay the current starts the motor, which opens the draft on the boiler in Fig. 166, thus increasing the air supply to the fuel bed and causing an increase in the heat output. Similarly, a slight increase of temperature above the established point causes a contact to be made between 11 and 6, and the motor is started, closing the drafts. The temperature for which the thermostat is set can be changed by moving the dial, which shifts the position of cam 4.

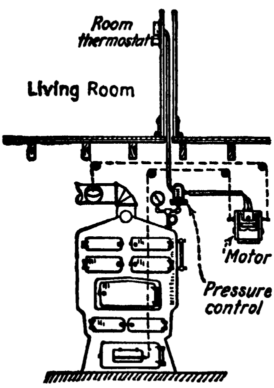

Fig. 166.—Method of connecting thermostat.

A simple form of compressed-air thermostat is shown in Fig. 167. The composition tube 3 is the temperature-sensitive element, and its expansion and contraction vary the rate of air leakage from the nozzle 8 through the simple arrangement of rods and levers (4, 5, and 6). The method of connecting to the secondary element is shown in Fig. 168. The varying amount of leakoff changes the pressure in the branch line because of the restrictive orifice in the supply pipe, and this changes the opening of the damper, or valve, by the compression of a flexible bellows, or diaphragm. Instead of a thermostat, a humidistat or pressure element may be similarly used.

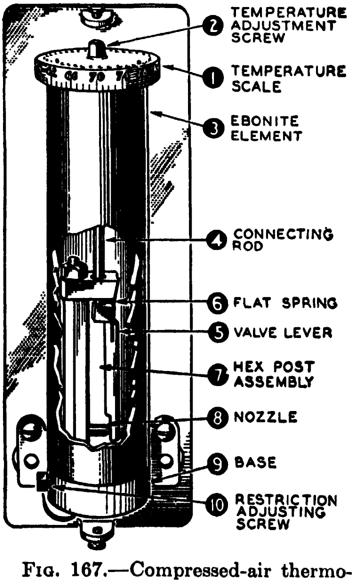

Fig. 167.—Compressed-air thermostat. (*Minneapolis-Honeywell Regulator Company.*)

The primary element for the control of humidity usually consists of a hygroscopic material such as hair or cross-grained wood. One form, designed for use in an electrical control system, is shown in Fig. 169. Elements for the control of pressures such as exist in air ducts usually consist of a large diaphragm or a float chamber.

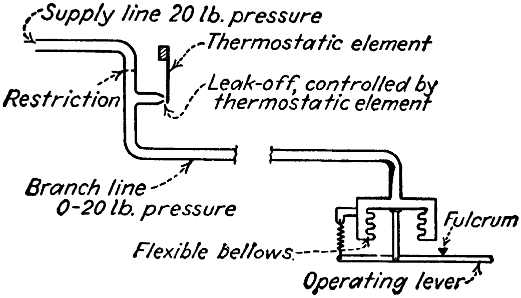
Supply line 20 lb. pressure
Thermostatic element
Leak-off, controlled by thermostatic element
Restriction
Branch line 0-20 lb. pressure
Fulcrum
Flexible bellows
Operating lever

FIG. 168.—Diagrammatic arrangement of compressed-air thermostat system.

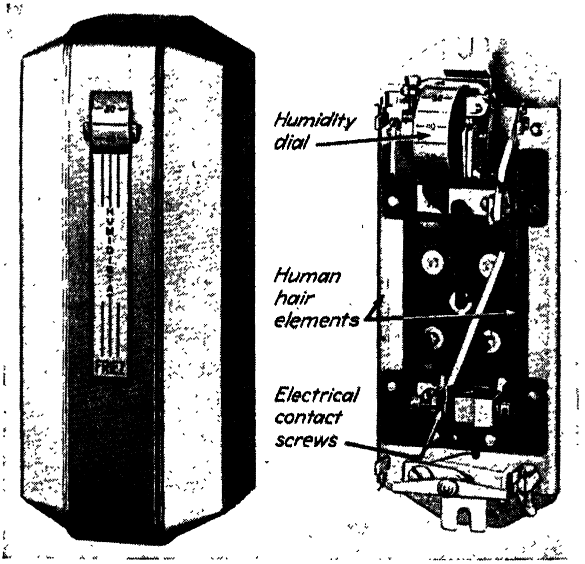

FIG. 169.—Humidity controller (with and without cover removed). (*Friez Instrument Division, Bendix Aviation Corporation.*)

192. Electrical Systems.—The electrical thermostat of Fig. 165 is a simple on-and-off type of control. In many cases a graduated type of control is called for, in which the damper or valve may assume intermediate positions. This requires a more complex type of control, one form of which is shown in Fig. 170. It is essentially an application of the common Wheatstone-bridge circuit.[1] The damper or valve controlled by the power unit

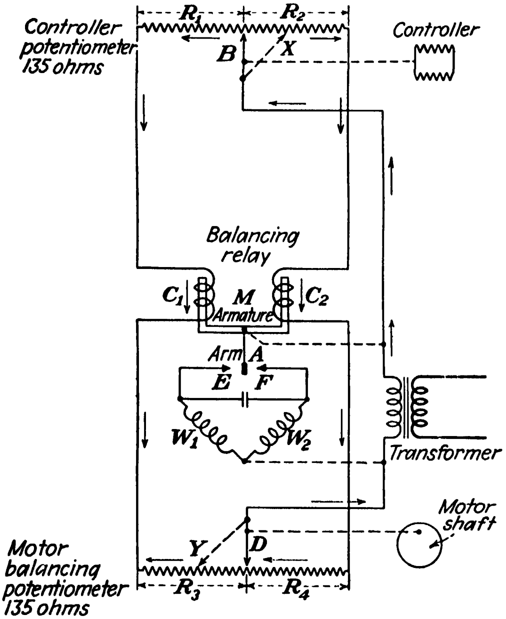

FIG. 170.—Diagram of one form of electrical control. (*Minneapolis-Honeywell Regulator Company.*)

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[1] The operation is as follows: Assume that the sliding contacts B and D are both in a central position as shown. Under these circumstances resistances R_1, R_2, R_3, and R_4 are all equal, and the current flowing in coil C_1 of the balancing relay is equivalent to that flowing in coil C_2. The U-shaped armature M will then be acted on by equal magnetic forces from coils C_1 and C_2, and the contact arm A will be floating in the gap between contacts E and F. Since no power is being furnished to either winding of the motor, the circuit to the motor windings is shown dotted, and under such conditions the motor will be at rest.

Any deflection of the controller will cause the sliding contact B to move toward some position such as X, which immediately unbalances the electrical system, as resistance R_2 will then be less than R_1. The net result is that more current will flow through relay coil C_2, and it will tend to follow the path of least resistance. Armature M will then move arm A to make takes a position that is directly related to the deflection of the primary element of the control instrument.

193. Secondary Control Elements.—In a compressed-air control system the secondary element is a diaphragm or bellows in which the air pressure is opposed by a coil spring. A radiator valve of this type is shown in Fig. 171.

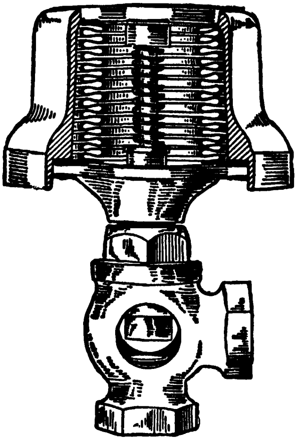

Fig. 171.—Radiator valve for compressed-air system of temperature regulation.

The valve is closed when air under sufficient pressure is admitted to the space surrounding the corrugated metal bellows. When the air pressure is released, the spring forces the valve open. If a pressure less than that required to close the valve exists around the bellows, the valve will take an intermediate position depending on the amount of that pressure.

A similar bellows, operating a louver damper, is shown in Fig. 172.

In electrical systems the secondary element is a specially designed electric motor unit with built-in gears to give high torque and slow motion.

194. Characteristics of Controls.—A control system may function in an on-and-off manner, or it may have a graduated action. In the latter case the change of position of the secondary element takes place gradually as the primary element is affected by the change of state of the controlled fluid—that is, by its change of pressure, temperature, or humidity.

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contact with *F*, and the current will be fed directly to motor winding W_2, which will cause the motor to run in the corresponding direction.

Blade *D* is linked directly to the shaft of the motor. As the motor shaft starts to rotate, it moves blade *D* of the balancing potentiometer to a new position such as *Y*. Now the right-hand side of the motor-balancing potentiometer will have a resistance value identical to the left-hand side of the controller potentiometer, which means that R_1 equals R_4 and likewise R_3 equals R_2. When this condition has been reached, the current flowing through both legs of the circuit will again be identical, and balancing relay coils C_1 and C_2 will have an equal effect on the armature. Arm *A* will then cease to make contact with *F* and the motor will again come to rest.

Positive (on-and-off) action is suitable for safety controls and for a few other applications, but most heating and air-conditioning systems require graduated controls. A graduated control usually gives much closer regulation than a positive control.

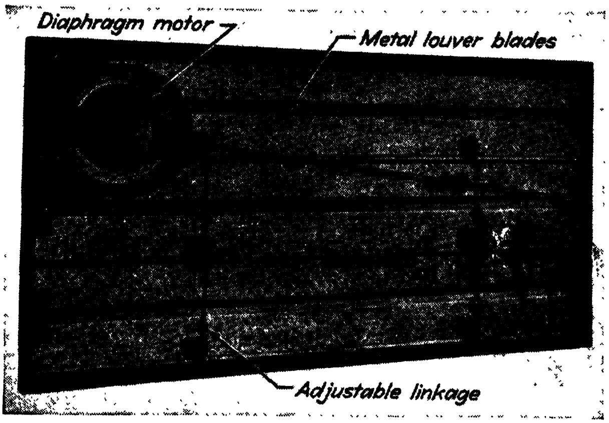

FIG. 172.—Air-operated louver damper. (*Powers Regulator Company.*)

There is a fundamental principle regarding graduated controls that should be borne in mind. A graduated control as ordinarily designed cannot maintain an absolutely constant state of the controlled fluid—that is, the air or water in the system—at varying loads. By its very nature it requires a small change of state in order to function. In Fig. 173, suppose that a control is set to maintain a temperature such as *A*. The valve or damper, to move from one extreme position to the other, requires a change of temperature from *B* to *C* along the line *B*-*C* (which may or

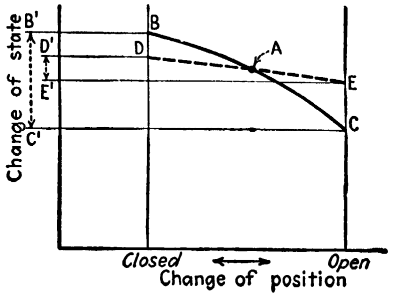

FIG. 173.—Characteristics of a control system

may not be a straight line). Therefore, between the no-load and full-load positions of the valve or damper the temperature will vary by an amount represented by $B' - C'$. In some controls the sensitivity can be adjusted. For example, it might be possible to give the control the characteristic *D-E*, in which case the variation in temperature would be only $D' - E'$. However, if a control is too sensitive, it is inclined to be unstable and "hunt"—that is, swing far above and below the control point. The choice must therefore be made between stability with greater variation of temperature on the one hand and smaller variation with possible hunting on the other.

The tendency to hunt and the violence and timing of the hunting depend largely on the lag in the system—that is, the length of time required for a movement of the control to change the conditions surrounding the primary element. If the lag is great, the primary element may cause a further change before it begins to be affected by the initial change.

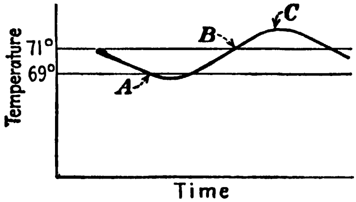

FIG. 174.—Inertia of heating system controlled by simple thermostat.

The remedy for hunting is usually either to reduce the sensitivity of the control or to change the position of the primary element to a point where it is more quickly affected by a change of conditions.

An example of time lag is the trouble often experienced with the simple thermostat, particularly when it is controlling a boiler using solid fuel. This is a form of hunting but with a long time cycle. In Fig. 174, when the temperature falls to the level at which the low-temperature contact functions, as at point *A*, the dampers open and combustion is accelerated, though not before the temperature has further fallen a slight amount. At point *B* the dampers close, but the inertia in the system, from the fuel bed, the hot radiators, and the steam in the piping, causes heat to be supplied for some time after the dampers close, with the result that the temperature rises to level *C*, which may be 4 to 6° above the temperature for which the thermostat is set, before it again starts to fall. This common occurrence, known as "overshooting," is a factor that must be reckoned with and not ignored.

It is less pronounced with boilers using oil or gas and with heating systems using lightweight convectors rather than cast-iron radiators. It is less noticeable in cold than in mild weather.

Several expedients have been devised to remedy this trouble. One is to add an electrical heating element to the thermostat itself, which cuts in when the low-temperature contact is made and, by artifically heating up the thermostatic element, causes the high-temperature contact to close and check the combustion rate sooner than it otherwise would (see Fig. 175).

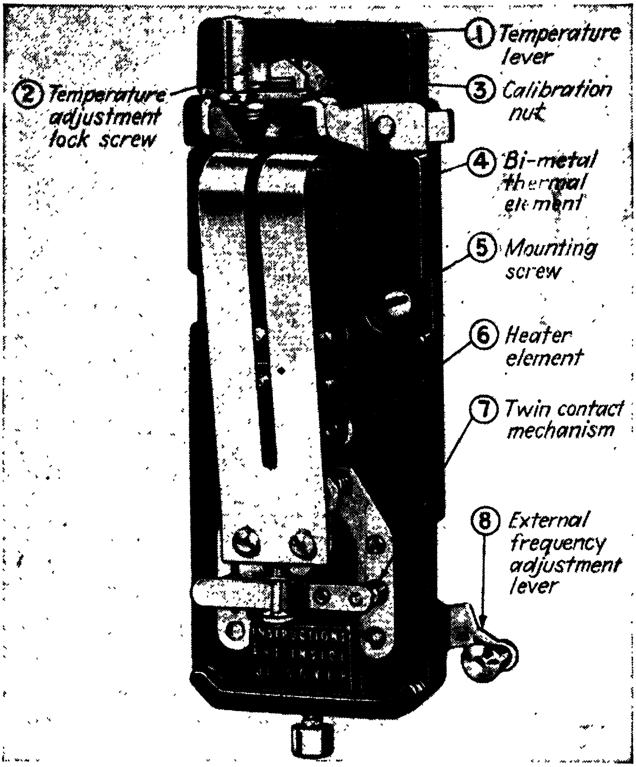
① Temperature lever
② Temperature adjustment lock screw
③ Calibration nut
④ Bi-metal thermal element
⑤ Mounting screw
⑥ Heater element
⑦ Twin contact mechanism
⑧ External frequency adjustment lever

FIG. 175.—Thermostat with heating element. (*Perfex Corporation.*)

195. Location of a Thermostat.—The most that can possibly be expected of any thermostat is that it maintain a constant temperature at the point where it is placed. Therefore the thermostat should be so placed in the room that a constant temperature maintained at that point will give the best results throughout the room with different outside temperatures. The full significance of this will be appreciated when it is remembered that at no time are the temperatures exactly uniform at all points of a room. There is a gradient from floor to ceiling and from the inner walls to the outer walls, and these gradients are both more pronounced when the outside temperature is low.

Many people now believe that an elevation of 2 to 3 ft. from the floor is more satisfactory for a thermostat than the usual height of 5 ft. In most cases an inside partition is probably a better location than an outside wall, although there are some advocates of the latter. Even some point near the middle of the room might be more suitable, but thermostats are not now designed for other than wall mounting. This question of thermostat location is far from being finally answered.

The simple form of thermostat is often used to control the temperature not only throughout a room but throughout an entire building. The degree of accuracy with which this can be done depends upon three factors: (1) the selection of the proper location of the thermostat, (2) the ability of the heating system to deliver to each room its proper quota of heat, and (3) the relative rates of cooling of the various rooms.

196. Effect of the Rate of Cooling.—The steam and warm-air systems, and to a lesser degree the hot-water system, when controlled by a central thermostat have essentially an *intermittent* action; that is, the rate of heat delivery is either very large or almost zero, according to whether the thermostat is or is not calling for heat. In gas- or oil-fired systems this difference is greater than in systems using solid fuel. Consequently, if one room cools more rapidly during the off period than does the room in which the thermostat is located, then the temperature in the former room may fall to uncomfortable levels before the next heating period. Thus a sun room with much window area, a third-floor bedroom, or a room over a cold garage may be chronically underheated, even though the radiator surface is adequate. It is a matter of *cooling rate* rather than of the actual amount of heat loss.

The best remedy is to arrange the heating system with a separate supply to such rooms, with separate temperature control. This is difficult to do once the heating system is completed, and the need should therefore be anticipated.

197. Temperature Control in a Hot-water System.—The hot-water system has certain characteristics that must be taken into account when considering temperature control. One of them is the high thermal capacity of the water in the system. Although this renders the heating effect less fluctuating than with the steam system, there is danger of overheating because of the large storage of heat in the radiators. The remedy is to lower the water temperature in mild weather. If, for example, a gravity system requires 180° water temperature when the outside temperature is zero, then when the outside temperature is, say, 40°, the water temperature should be approximately 140°. In fact, a rough rule is that the *sum* of the water temperature and prevailing outside temperature should be about 180°. There are devices, such as the one described in Chap. X, Art. 184, for automatically changing the water temperature with relation to the outside temperature in such a manner, and when such a device is supplemented by a simple thermostat a very satisfactory result can be obtained. The control of the water temperature may be accomplished fairly well by hand if the controller is reset periodically through the winter as the weather changes.

The foregoing applies particularly to a small system, as in a residence. In large buildings where there is an attendant on duty, the water temperature can be changed by hand adjustments of the temperature controller on the water heater, which may be a draft controller if a fuel-fired boiler is used, or a control applied to the steam-supply valve if heat exchangers supplied with steam are used.

The hot-water system readily permits adjustment of the heat supply, provided the radiators are correctly proportioned.

198. Self-contained Thermostatic Radiator Valves.—In the effort to obtain for individual rooms a thermostatic control that would be less costly and easier to install, a number of thermostatic radiator valves have been brought out. A thermostatic member is incorporated in, or directly connected to, the valve. The thermostat usually consists of a bulb containing a volatile liquid that expands as the temperature increases, so as to close the valve, and vice versa. These valves are all direct-acting; that is, the force required to open and close the valve is developed by the thermostatic element instead of by some external source of power such as compressed air or electricity.

The problem in the design of such a valve is to locate the thermostatic element so that it will not be affected by the heating element and will at all times be in the path of a current of air whose temperature is truly representative of the room air. With ordinary cast-iron radiators the usual arrangement is as shown in Fig. 176, where the sensitive bulb is an integral part of the valve. In this type of application the thermostatic element of the valve is subjected to radiant heat from the supply riser, and consequently it will not maintain so accurate a control as can be secured with the remote-bulb arrangement as shown applied to a convector in Fig. 177.

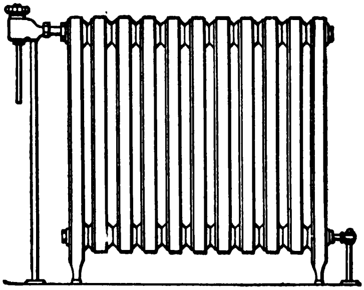

FIG. 176.—Self-contained thermostatic radiator valve on ordinary radiator.

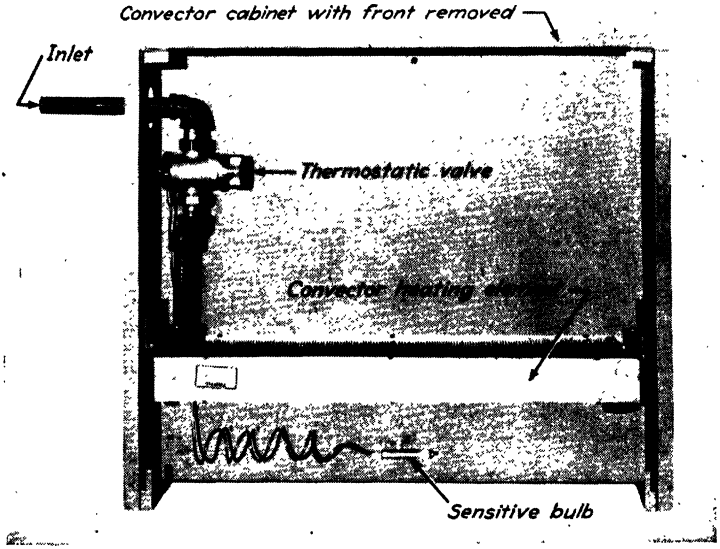
Convector cabinet with front removed
Inlet
Thermostatic valve
Convector heating element
Sensitive bulb

FIG. 177.—Self-contained thermostatic radiator valve controlling a cabinet convector. (*Trane Company.*)

A cross-sectional view of one design of self-contained thermostatic valve with remote-control bulb and capillary tube is shown in Fig. 178. Air passing over the sensitive bulb causes the volatile liquid to expand and contract in the capillary tube, which actuates the thermostatic bellows in the valve, thus varying the valve opening with air temperature. A hand-controlled adjusting screw on the opposite end of the valve stem is provided to vary the temperature setting of the valve. With this type of regulation steam is throttled so that the amount passing through the valve is just sufficient to heat the radiator fractionally to the extent required to maintain the desired room temperature. Valves of this type are obviously not applicable to one-pipe radiators because they have a throttling action.

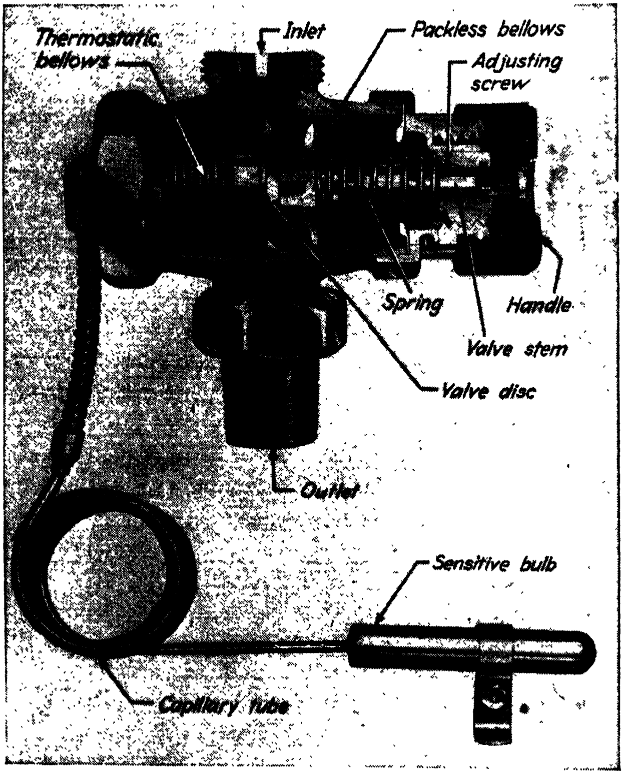

FIG. 178.—Cross-section of thermostatic radiator valve. (*Trane Company.*)

One difficulty with the operation of thermostats that are under control of the occupants is the apparent impossibility of making the average person understand how a thermostat is supposed to function. People are apt to change the adjustment from "hot" to "cold" without taking the trouble to find the intermediate position at which the thermostat can be left to do the regulating. This behavior is due in some cases to improper functioning of the thermostat. A thermostat which is too sluggish, on the one hand, or which, on the other, suddenly opens the valve wide and fills the entire radiator with steam on a mild day is sure to cause uncomfortable conditions, and the occupant becomes discouraged in attempting to regulate it.

In a large room occupied by many people the only satisfactory type of thermostat is one in which the adjustment is locked and can be changed only by the authorized person.

199. Orifice Control.—Although individual room thermostats of any of the types that have been described are in one sense the ideal method because they are supposed to satisfy exactly the needs of each room independently, they are not in common use. Their cost is high, especially to install them in existing buildings, and they require maintenance and adjustment.

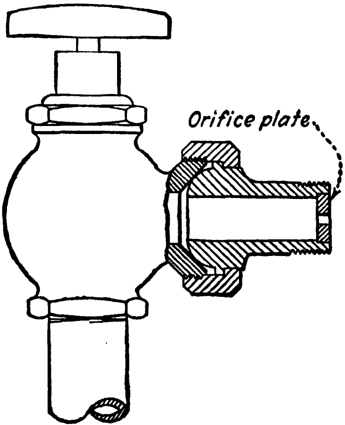

FIG. 179.—Radiator valve with orifice.

Temperature control usually results in marked economy of operation as well as in increased comfort, and in order to secure such economy a number of methods of control have been devised which are relatively cheap to install and which save heat, although they do not usually give very exact temperature regulation. They usually take the form of centralized controls as contrasted with the individual room control.

The hot-water system, as has been explained, is readily adaptable to such control by merely varying the water temperature, but no such adjustment is possible in a steam system because one of the unfortunate properties of steam is that its temperature is not changeable over a wide range. However, one quite successful expedient is to install an orifice at the inlet of each radiator, of such size that when the steam pressure in the supply piping is varied over a definite range, the radiators will be fractionally heated to the desired degree. A schedule of the pressure to be carried for a given outside temperature is used. This form of control, as is the case with all those that do not have a thermostat in each room, must be supplemented by hand control of the individual radiators.

The orifice may be placed at any one of several points in the radiator supply pipe. Figure 179 shows an orifice plate in the tailpiece of the supply valve. It is simply a brass disk, $\frac{3}{32}$ in. thick, pressed into the opening. Orifices can be used only on a two-pipe system, because in a one-pipe system the condensation could not find its way out against the high velocity of steam through an orifice.

The proper diameter of orifice for radiators of various sizes is given in Table 59, which is based on a pressure drop of 6 in. of mercury across the orifice when the full amount of steam necessary to supply the radiator is flowing. The pressure differential would then be varied between 0 and 6 in. of mercury, depending upon the outside temperature as given in Table 60.

TABLE 59.—SIZE OF RADIATOR ORIFICES[1]

(Orifice plate $\frac{3}{32}$ in. thick. Maximum pressure differential 6 in. Hg)

Orifice diameter, in.	Size of radiator, sq. ft.	Orifice diameter, in.	Size of radiator, sq. ft.
$\frac{3}{32}$	18	$\frac{3}{16}$	61
$\frac{7}{64}$	24	$\frac{13}{64}$	70
$\frac{1}{8}$	30	$\frac{7}{32}$	80
$\frac{9}{64}$	37	$\frac{15}{64}$	91
$\frac{5}{32}$	44	$\frac{1}{4}$	102
$\frac{11}{64}$	52		

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[1] From SANFORD and SPRENGER "Flow of Steam through Orifices into Radiators," *Trans. A.S.H. & V.E.*, 1931.

Some designers advocate placing orifices in the steam mains as well as at the radiators. In the authors' opinion, however, if the mains are adequate in size it is better to have a fairly high pressure drop through the orifices, as given in Table 60, and use orifices only at the radiators.

TABLE 60.—REQUIRED PRESSURE DIFFERENTIAL ACROSS ORIFICE FOR DIFFERENT OUTDOOR TEMPERATURES[1]

(Atmospheric pressure in radiator; room temperature 70°)

Outdoor Temperature, °F.	Pressure Differential, In. Hg
0	6.0
10	4.5
20	3.2
30	2.1
40	1.2
50	0.5
60	0.1

[1] From SANFORD and SPRENGER, "Flow of Steam through Orifices into Radiators," *Trans. A.S.H. & V.E.*, 1931.

The orifice method is inexpensive to install and usually produces important savings in steam consumption. It requires frequent attention for the readjusting of the pressure and is therefore suitable only for buildings in which an intelligent operator is on duty. There are, however, control systems available for use with orifices, in which the differential pressure between the supply and return pipes is automatically controlled with respect to the outside temperature.

200. Temperature Control by Varying the Steam Pressure.—Although the temperature of steam cannot be varied over a sufficient range to give complete control of the heat supply, a certain degree of control can be obtained by varying the steam pressure. This is practicable only in a vacuum system. At 1.0 lb. pressure the steam temperature is approximately 215°; at 22 in. vacuum it is approximately 152°. If the room temperature is 70°, the heat output of the radiator full of steam at 22 in. vacuum is about 45 per cent of that when it is full of steam at 1.0 lb. pressure. Of course it is necessary that the pressure *in the radiator* be varied, and not merely the vacuum in the return system. This is done by throttling the main steam supply or reducing the fuel-burning rate until the proper pressure is reached. Vacuum pumps of large air capacity are required for obtaining high vacuum.

201. Zoning.—Another way of securing partial temperature control and its accompanying saving in steam consumption is to divide the building into zones to which the steam supply can be cut off or turned on by valves remotely controlled from a central operating point. The zones are chosen with regard to their hours of occupancy and exposure to the sunshine. Thus heat is not supplied to a zone when it is unoccupied, and in mild weather it is supplied only to rooms on the shady or windward side of the building. Zoning is justified only in rather large buildings but when properly applied is productive of considerable savings. It is sometimes combined with other methods of control.

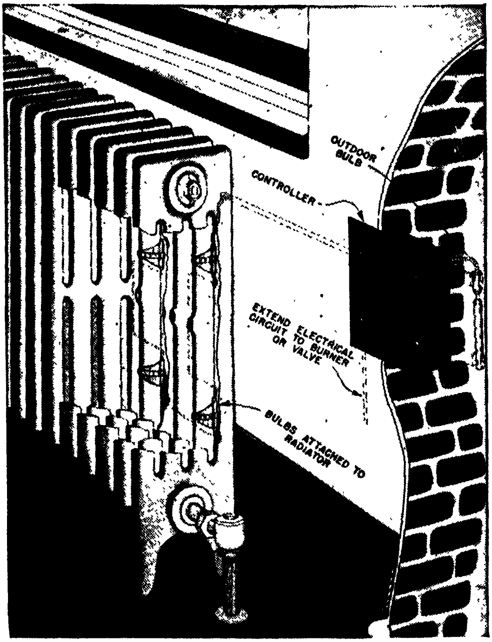
OUTDOOR BULB
CONTROLLER
EXTEND ELECTRICAL CIRCUIT TO BURNER OR VALVE
BULBS ATTACHED TO RADIATOR

FIG. 180.—Outside weather and radiator control. (*Johnson Service Company.*)

202. Control from Outdoor Temperature.—In recent years a new conception of heat control has arisen, based rather more on the desire to economize in the use of steam than to secure perfect temperature control. In these newer controls the outside temperature instead of the inside temperature is used as a source of control, or in some cases the two are combined. The heat supply is thus governed primarily by the outside rather than the inside temperature. The orifice method, previously described, is usually operated in this manner, and a variety of designs of automatic controls have been developed, some using orifices and others feeding steam intermittently into the system.

Intermittent control is usually accomplished by correlating the temperature of a typical radiator in the building with the outdoor temperature. One method is shown in Fig. 180. A thermostat having two opposed temperature elements is used. One bulb is placed outdoors, and the other four averaging bulbs are placed on a properly selected radiator. The switch controlling the power supply to the fuel burner or steam valve is affected by the combined action of the temperature bulbs; and when the outside temperature falls, the radiator bulbs must maintain a higher temperature so that their combined effect will be approximately constant. To satisfy this condition the switch causes the heat supply to function intermittently.

The same device can be used to control the temperature in a hot-water heating system.

These central control systems fall into one or the other of the following groups: (*a*) those in which the operating is controlled by the attendant with reference to the outside temperature and (*b*) those in which the operation is controlled directly by the outside temperature.

203. Summary.—The several methods of heating-system control may be classified according to type, as follows:

Type of Control	Remarks
Central thermostat	Standard method for residences and small buildings
Individual room thermostats Compressed-air type Electrical type Self-contained radiator valves	Used principally in public buildings
Central control With radiator orifices By varying pressure only Intermittent supply	Gives approximate temperature regulation in large buildings. Usually installed for sake of economy
Zoning	Very rough temperature control but often justified from standpoint of economy

The central thermostat, supplemented by pressure control in the case of steam and by temperature control of the heating medium in the cases of warm-air and hot-water systems, is the almost standard method for residences and other small buildings.

Individual room thermostats of the compressed-air, electrical, or self-contained type are used in public buildings, especially the first two mentioned. They give, theoretically, true temperature control but are costly and not always satisfactory in performance.

The other forms of control that are operated from a central point—orifice systems, pressure variation, and intermittent systems—are usually installed to reduce heating costs and give only approximate temperature control. Zoning gives a still less exact temperature control if used alone but often supplements other forms.

Some idea of the savings that can be effected by the use of control equipment can be observed from Table 61. These buildings were all supplied with steam from district-heating systems in various cities. The figures should not be regarded as a comparison between the different types of control because of variables in the buildings and occupancy, but they show conclusively the savings produced.

TABLE 61.—COMPARISON OF STEAM CONSUMPTION BEFORE AND AFTER INSTALLING TEMPERATURE-CONTROL EQUIPMENT

Type of control	Steam for heating, lb.			
	No. of bldgs.	Per degree day		Saving, per cent
		Before	After	
1. Subatmospheric pressure system	5	1,720	1,450	15.7
2. Orifice system controlled by an outdoor thermostat	2	2,039	1,600	21.6
3. Orifice system controlled by group of outdoor thermostats	1	4,580	4,255	12.2
4. Orifice system—manual control	3	7,930	6,750	14.9
5. Orifice system—electric zone control	2	4,110	4,025	2.1
6. Intermittent control—electric timer	5	1,340	1,200	10.4
7. Intermittent control actuated by outdoor and radiator temperatures	14	772	623	19.3
8. Central thermostat	19	776	585	24.6

204. Shutting Off Heat at Night.—The shutting off of heat at night is a most important factor in economical heating. Consider the load curves of Fig. 181, which were taken from an actual building supplied with steam from a district heating system. The total amount of steam used during the 24-hr. period when the heat was shut off during the night hours was 48 per cent less than when it was not shut off. The slightly higher consumption during the morning hours is more than balanced by the saving during the night. This result would be expected from a consideration of the fact that when the building is allowed to cool during the night, the average temperature difference above the outdoor temperature and consequently the total heat consumption are less. Of course the amount of radiator surface must be ample to warm the building quickly in the morning.

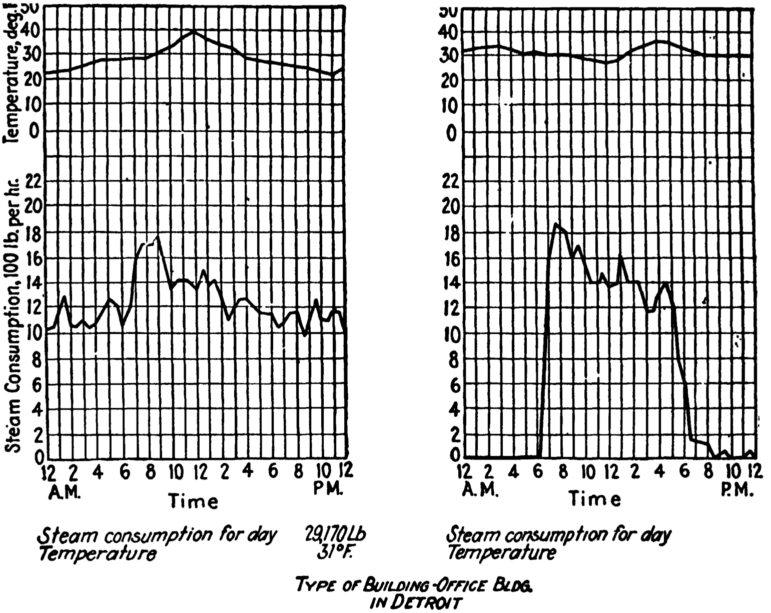

FIG. 181.—Load curves showing the saving in steam due to shutting off heat at night.

Problem

1. Calculate the thermal content above the room temperature of the radiator (plus the fluid which it contains) for the following three conditions, assuming that the radiator is designed for a maximum heat output of 13,000 B.t.u. per hr. with the room temperature at 70°: (*a*) Hot-water radiator, average water temperature 185°, cast-iron radiator of the ordinary tube type; (*b*) steam radiator of same type with steam at 215°; (*c*) steam convector of which total weight is 70 lb. and the internal volume 145 cu. in. Convector is constructed of copper throughout. Assume for the cast-iron radiators a volume of 28 cu. in. and a weight of 5½ lb. per equivalent sq. ft. of surface.

CHAPTER XII

CENTRAL AND DISTRICT HEATING—ELECTRIC HEATING

205. Central Heating.—The term "central heating" is somewhat ambiguously used. In Europe it refers to the heating of buildings from a single source, located within the building, as differentiated from grates or stoves in the separate rooms. In America the term refers to the heating of several buildings from a central plant, such as a group of institutional buildings, and is sometimes used with reference to the heating of portions of a city. The latter, however, is more properly termed "district heating."

The central heating plant for a group of institutional or industrial buildings is usually superior to separate heating plants in the individual buildings and is the type more frequently installed. It is more efficient in fuel consumption, requires less labor, and is often cheaper to install.

The material in this chapter does not deal with the design of the plant itself nor with possible combinations with electrical generation but with the manner of distributing the heat supply. In many institutions and in some factories the hot-water heating system is found to be very satisfactory for a central plant. It permits the adjustment of the amount of heat sent out from the plant and thus results in more generally comfortable heating conditions. It can be arranged to utilize the heat exhausted from electrical generating units through the use of either heat exchangers, in which the water is heated by steam, or surface condensers, which operate at a low vacuum and circulate the water through them.

There are two methods used for taking care of the underground pipes that carry the steam or hot water. They may be installed in tunnels large enough for men to walk in, or they may be merely enclosed in a protective conduit. For institutions particularly, the tunnel system has many advantages. It is not extremely costly if it can be built by the cut-and-cover method instead of by tunneling. Another advantage is that there are usually other pipes, carrying high-pressure steam or hot water for lavatories, and perhaps electric cables, which may be housed in the same tunnel. Ready access to the pipes for inspection and repairs is of course another advantage of the tunnel method.

The principles involved in the design of central heating systems for institutions and of district heating systems are substantially the same, and the same sort of special appliances are used.

206. Commercial District Heating.—There are in many of our larger cities extensive systems that distribute heat to buildings as a form of public-utility service. Although not a particularly profitable form of undertaking during its early years, district heating has recently been more favorably regarded, and when developed in suitable territories it fills a real demand and can be financially successful. Its most important advantages to the building owner are the freedom from the dirt and nuisance of operating his own plant and the release for rental or other purposes of valuable space that otherwise would be occupied by boiler plant and chimney. The most suitable field is in the business districts of cities where a large quantity of heat can be distributed with a relatively moderate investment in underground pipes.

In this country steam is the medium almost universally used for conveying the heat. Hot water was formerly used to some extent, and a few plants are still in operation, but hot water has several disadvantages in comparison with steam. Its inability to be readily metered is a great disadvantage in a public-utility service, and it cannot be used in tall buildings unless an extremely high pressure is carried. The following discussion relates principally to steam systems.

207. Efficiency.—It is interesting to compare the relative overall thermal efficiencies of the district heating system and the individual boiler plant. The steam-generating plant of the district heating system will be more efficient than the individual boiler plant, but there are other losses to be considered, such as the heat loss from the distribution system, steam leakage, and loss of heat in the condensate drained from the radiators.

The efficiency of a modern steam-generating station will average from 75 to 85 per cent. The efficiency of distribution is from 80 to 90 per cent, and the efficiency of utilization of the heat within the buildings will be at least 90 per cent. (The remainder is carried away with the condensation.) Assuming 80 per cent for the steam-generating station, 85 per cent distribution efficiency, and 93 per cent efficiency of utilization, an over-all efficiency of 63 per cent is obtained. This may be compared with 50 per cent, which is probably about the average efficiency of a small individual boiler plant, and 60 per cent, which is probably representative of a large individual plant.

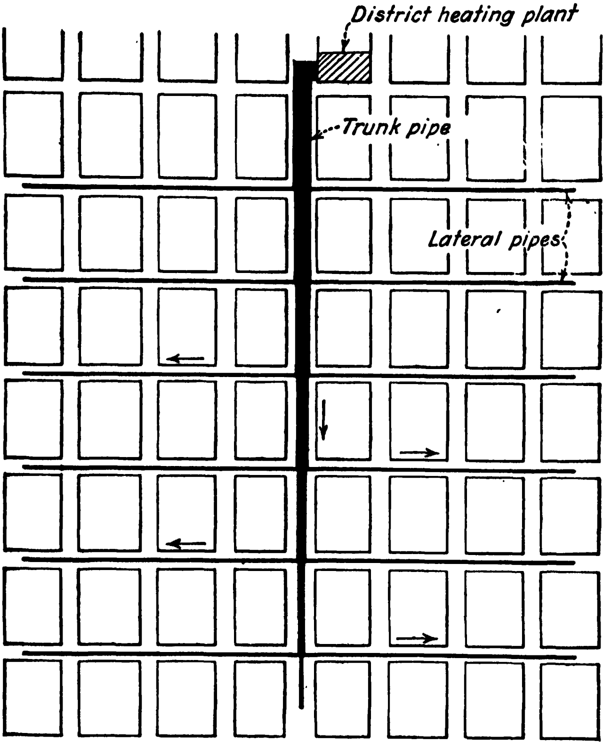

FIG. 182.—Trunk system of steam distribution.

In many district heating plants the steam is passed through electrical generating units before being distributed for heating, thus making possible the generation of considerable amounts of by-product power at a very low fuel cost. The increasing use of higher boiler pressures increases the theoretical possibilities in this respect. In some cases, however, it appears to be more expedient from a commercial standpoint to pass the steam direct from the boilers to the distribution pipes.

208. Methods of Steam Distribution.—The distribution pipes in a large system constitute a grid or network from which service connections are taken to the individual buildings. There are two methods of distribution, which are illustrated in Figs. 182 and 183. The trunk system of Fig. 182, which is the more common, employs a central trunk or stem, which is of large diameter near the plant and decreases in diameter toward its remote end. The feeder system of Fig. 183 makes use of feeder pipes that extend from the plant to strategic points in the distribution network. No consumers are ordinarily supplied directly from feeders, which carry steam at high velocities (up to 50,000 ft. per min.) and with rather large pressure drops, resulting in

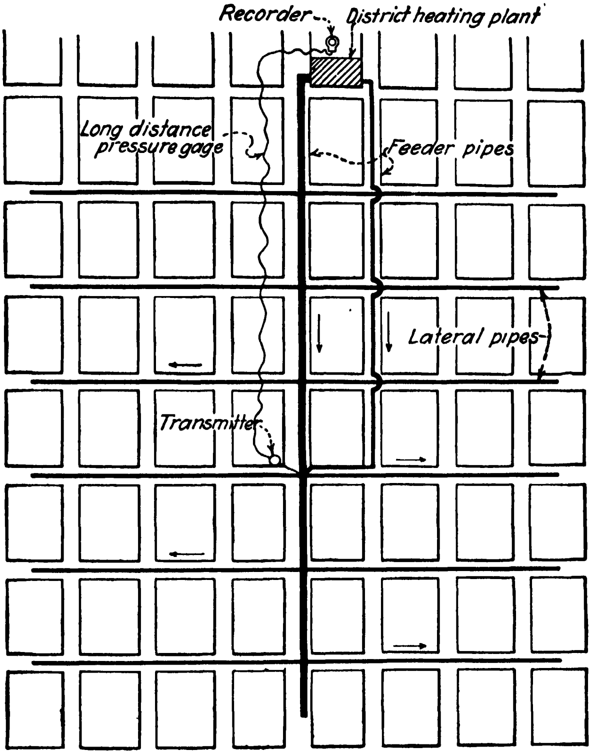

Fig. 183.—Feeder system of steam distribution.

relatively small pipe sizes. The pressure is maintained nearly constant at the remote end, or feeding point, as indicated by a long-distance gage (electrically operated), and the pressure carried in the feeder at the plant is adjusted according to the steam demand so as to maintain a constant pressure at the remote end. The feeder system is a convenient method of increasing the capacity of a network that has become inadequate, as additional feeders can be installed to new centers of load. The trunk system, however, is used in most cases.

The steam pressures carried in distribution networks vary greatly in different cities. If low-pressure steam, exhausted from electrical generating units, is utilized, the pressure may be from 5 to 10 lb. gage. Restaurant cooking apparatus, however, requires a pressure of 10 to 25 lb., and this range is often chosen. A reducing valve is then used in each building to maintain a suitably low pressure for building heating, the full street pressure being used for the cooking.

Pipe sizes are determined from a consideration of present and probable future load. The sizes should be selected on the basis of friction loss rather than velocity or any other criterion. The use of the full available pressure drop results in relatively small pipes and a consequently lower installation cost, which is quite important.

The chart in Fig. 184 is a graphical solution of the Unwin pressure-drop formula, which is generally used in district heating work. Its pipe sizes are somewhat liberal.[1]

In laying out a distribution system it is first necessary to estimate the maximum demand of each building and then to compute the quantities of steam flowing, with due regard for possible future loads. When the piping is arranged in a network, the flow through the different lines is often complex, and several trial calculations are required to bring out the true conditions of flow and pressure drop.

209. Underground Steam Pipes.—There are four major points to be considered in the design of underground steam piping, namely, (1) heat insulation, (2) protection from mechanical injury and corrosion, (3) drainage of condensation, and (4) provision for expansion.

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[1] For allowances for fittings see Table 48, Art. 166, Chap. IX.

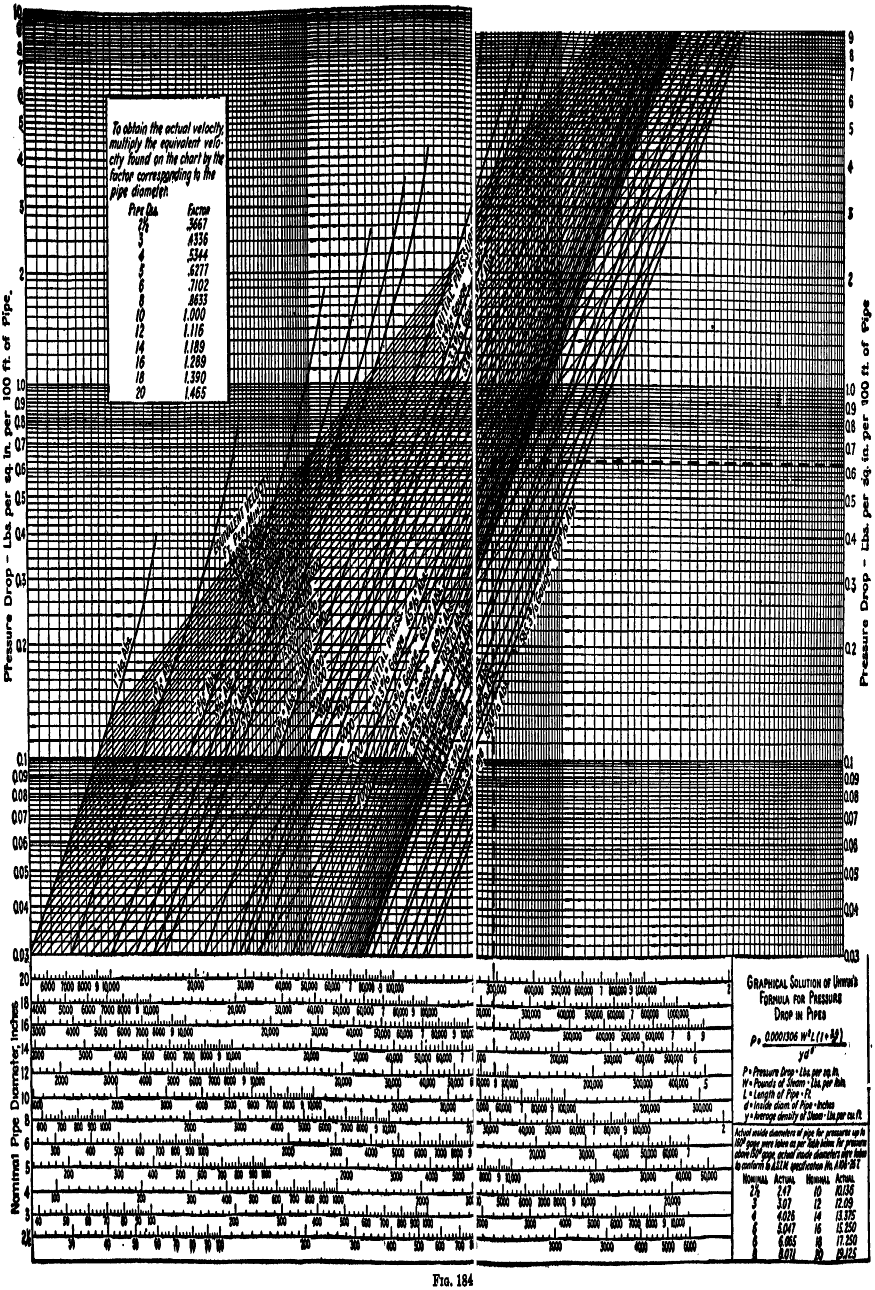

Fig. 184

[1] As an example of the use of the chart, assume a 12-in. pipe carrying 90,000 lb. of steam per hour at an av essure of 91-lb. gage. What will be the pressure drop per 100 ft. of pipe?

From the 90,000-lb. mark on the scale for 12-in. pipe at the bottom of the chart, proceed vertically to a point onding to 91 lb. as shown by the sloping lines. Thence proceed horizontally to the right-hand side and read the pressure drop 0.68 lb. per sq. in. per 100 ft. The velocity may be obtained by multiplying valent velocity, 7,200 ft. per minute, by the factor 1.116 taken from the table, giving a velocit- of 8,035 ft. per min.

The pipe is always enclosed in some sort of protecting conduit, which permits free expansion and is strong enough to withstand the pressure of the earth and any additional load that may be imposed. The insulation may be attached directly to the pipe or to the inner surface of the conduit, or it may be a loose material packed between the pipe and the conduit.

Several designs of underground construction are shown in Fig. 185.

Design *a* is a practically watertight construction, developed for very wet soil. It is not a commercial design and is not in very common use.

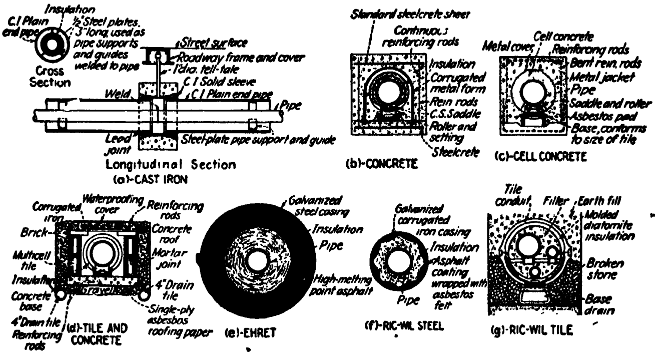

FIG. 185.—Various types of underground steam-conduit construction.

Design *b* is a common type used by several district heating companies with various slight modifications. It has the advantages of being simple and of being constructed from common, easily available materials. In this construction the bottom is poured first, and after the pipe is installed the sides and top are poured over the steel form.

Design *c*, a proprietary design, uses a very light concrete made porous by a gas-forming chemical. The porous "cell concrete" is the insulator and is poured directly around the pipe into a steel form.

Design *d* is one of many similar designs that employ masonry, tile, or concrete.

Designs *e*, *f*, and *g* are proprietary designs that can be understood by studying the accompanying notes. Designs *e* and *f* are factory assembled and the lengths are joined by welding, after which a special joint construction for the conduit and insulation is applied.

To carry off the ground water that would otherwise seep through the walls of the conduit, it is usually desirable to install a drain tile, as shown in some of the illustrations. The bottom of the trench is filled with coarse gravel or broken stone, in which are laid one or more runs of field tile or vitrified sewer tile (preferred), which carry the seepage water to a sewer or other drainage point.

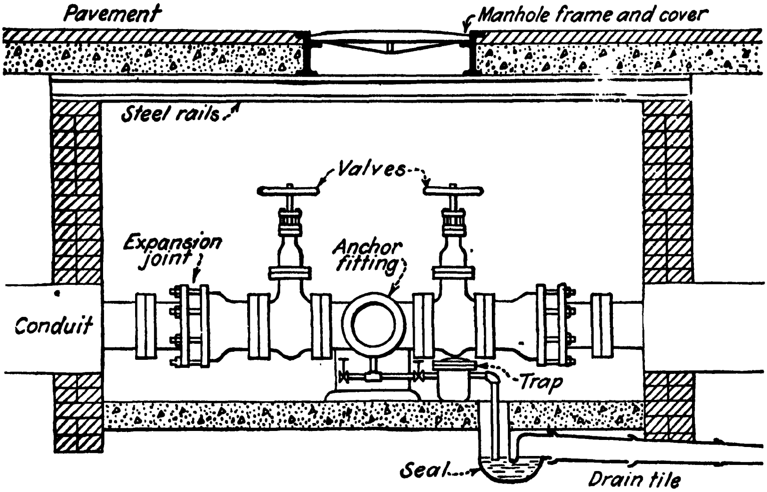

Fig. 186.—Manhole.

When the pipe is laid close to or below water level, it is useless to attempt to drain off the water, and reliance must be placed upon the watertightness of the conduit.

None of these conduits except design *a* could be called strictly watertight. It is very difficult to make a concrete or masonry duct absolutely and permanently waterproof, and it is better to be prepared for some seepage. The pipe itself may be protected by a jacket that can be made really waterproof, and provision can be made for draining off the seepage from the inside of the conduit.

Manholes are required at intervals to house valves, traps, and some kinds of expansion joints. Figure 186 shows a typical form of construction used in city streets.

210. Expansion Fittings.—In institutional work it is permissible to take care of the linear expansion of the pipe by permitting it to move freely and to bend the pipes connected at right angles to it. This method is not suitable for the longer distances encountered in district heating, where expansion joints must be provided at intervals. The slip joint of Fig. 187*a* is one popular form. This particular design has tie rods that prevent the joint from pulling apart. The permissible travel in a slip joint varies from 3 to 6 in., depending upon the size and the design. Slip joints are usually located so as to take care of 200 to 300 ft. of pipe.

The expansion joint of Fig. 187*b* utilizes the flexure of two corrugated copper diaphragms. This form of expansion joint will not take care of so much travel as the slip joint, and the diaphragms must be placed at intervals of approximately 60 ft.

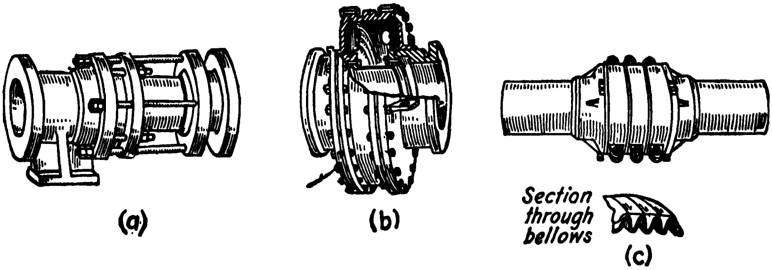

FIG. 187.—Expansion devices.

The joint of Fig. 187*c* consists of a corrugated copper bellows enclosed by split cast-iron rings that limit the flexure of each corrugation to a safe amount. The ends of this fitting are designed for welding into the pipe line. Unlike the slip joint, the fittings of types *b* and *c* need not be located in a manhole for accessibility.

211. Anchorage and Drainage.—It is always necessary to anchor the pipe midway between the expansion fittings. This is done by a clamp of some sort around the pipe or by a fitting with an anchor base. Branch connections are preferably made at or near the anchor points. If it is necessary to connect a branch at a point where there is considerable expansion movement, ample clearance must be provided around the branch pipe, and it must be quite flexible.

The pitch of underground pipes should be at least 1 in. in 50 ft., preferably in the direction of the steam flow. The condensation should be drained off, through traps, at intervals of not more than 30 ft. and from all low points where the pipe has an actual water pocket. The contour of the ground will often dictate the pitch.

The most suitable depth below the surface depends upon the nature of the installation. In industrial work with level ground, 2 or 3 ft. is sufficient, but in district heating work the average depth is usually about 6 ft. to the center line of the pipe. Fewer obstructions are encountered at the lower levels, and the heat loss is less because of the insulating effect of the soil. The pipe should consist of a series of perfectly straight sections between anchors and expansion joints.

212. Heat Loss.—Heat loss from a pipe buried in soil is somewhat different from that of a pipe in air, because of the insulating

TABLE 62.—SUMMARY OF TESTS OF HEAT LOSS FROM UNDERGROUND PIPES*

Test	*A*	*A*	*B*	*B*	*C*	*D*
Size of pipe	6 to 12 in.	6 to 10 in.	12 in.	8 in.	8 in.	4 to 20 in.
Length of pipe, feet†	....	....	301.5	681.25	576	19,302
Duration of test	8 months	8 months	2,376 hr.	2,376 hr.	8 months	8 months
Average steam pressure, pounds gage	5	25	38.6	38.6	26.4	2
Character of soil	Dry clay	Dry clay	Dry clay	Dry clay	Dry clay	Gravel, well drained
Type of conduit	Wood casing 81 per cent Concrete, 19 per cent	Wood casing 90 per cent Concrete, 10 per cent	Concrete	Concrete	Concrete	Wood‡ casing 90 per cent
Thickness of insulation, inches	1§	1§	1½	1	1	
Condensation loss, pounds per square foot of pipe surface per hour	0.0511	0.0588	0.0755	0.0816	0.0429	0.0585
B.t.u. loss per square foot of pipe surface	49.1	54.9	60.5	75.1	42.9	

A Test made by The Detroit Edison Company, 1913 to 1914.
B Test made by The Detroit Edison Company, 1923.
C Test made by The Detroit Edison Company, 1923 to 1924.
D Test made in Peoria, Ill.

* WALKER and CROCKER, "Piping Handbook."

† These lengths do not include the branch service connections to buildings.

‡ Balance, combinations of brick, hollow tile, and concrete, with 2-in. insulation.

§ In concrete conduit only—no other insulation in wood casing.

effect of the soil. The conductivity of the soil varies considerably, depending upon its composition and dryness, particularly the latter. The heat loss in wet ground will be greater (sometimes by as much as 200 per cent) than that from the same conduit in dry soil.

The heat loss from a pipe buried in the ground is not proportional to the external surface of the pipe. The loss per square foot of surface is much larger for small than for large pipes.

The depth to which a covered pipe is buried in the soil makes very little difference in the heat loss, provided the center of the pipe is 2 ft. or more below the surface of the ground. Beyond 2 ft., unless the pipes are very large, the heat loss remains substantially the same.

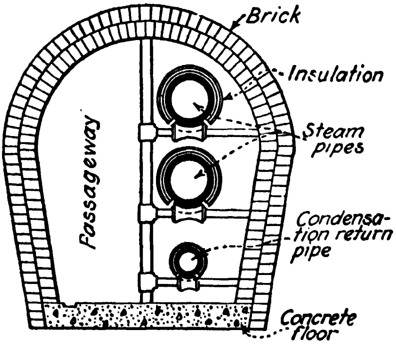

FIG. 188.—Tunnel used for district heating.

The economical thickness of insulation is difficult to determine exactly, except by actual test in existing soil conditions. The first layers of insulation are much more effective than succeeding layers, and the economical thickness may easily be exceeded.

The heat loss from the pipes in a district heating system is not so serious a matter as is generally supposed. If the pipe sizes are properly chosen, the amount of heat loss will be small in proportion to the quantity of steam flowing through the pipe. A well-designed district heating system delivers to the consumer from 80 to 90 per cent of the heat leaving the boiler plant, which compares favorably with other kinds of public-utility service. Table 62 gives the results of some actual tests.

213. Tunnels.—Tunnels of brick or concrete are sometimes used when several pipes are to be installed. The size and shape of tunnel used will depend upon the number of pipes, the character of the soil, and the depth of the tunnel in the ground. Figure 188 shows a tunnel of a design used in a large district heating system. It is sufficiently strong to withstand considerable earth pressure and is located at depths of 25 to 60 ft. below the surface.

The use of tunnels in commercial district heating is warranted only when two or more pipes are to be installed in the same street or when the subsurface conditions do not permit the installation of the conduit type of construction.

214. Return of Condensation.—In institutions it is quite desirable and usually perfectly feasible to return the water of condensation to the boiler plant, for the sake of salvaging the heat in the condensate, the cost of the water, and the cost of treating it chemically. If the contour of the ground and the location of the plant with respect to the buildings are favorable, a gravity system can be used, but sometimes it may be necessary to install pumps in the several buildings to force the water back to the plant.

In commercial district heating where a large number of buildings are served, it is not usually economical to return the condensation because of the difficulties of pumping and the high cost of the return piping. When it is done, the return pipe is usually carried in the same conduit or tunnel with the steam pipe. In a tunnel system a small condensation pipe is necessary to take care of the discharge from the traps that drain the steam lines.

215. Consumer's Piping.—As the steam pressure in the supply pipes is in most cases considerably above that which is desirable in the radiators, a pressure-reducing valve is required at the point of supply. The condensation is drained from the consumer's piping system through a float trap and passes through the meter and thence to the sewer. The arrangement recommended for a small system is shown in Fig. 189.

There are certain special considerations that govern the design of the piping when the steam supply is taken from the district heating mains. The amount of heat used in a building supplied from its own boiler plant depends upon the amount of fuel fed to the boiler and is therefore under direct control; but with district steam service the supply is virtually unlimited, and unless the heat is carefully controlled and shut off when not needed, there will be an expensive waste of steam. The design of the heating system must be such that the supply can be *easily* controlled.

Valves that should be operated frequently must be in accessible positions or provided with extension stems or chains. When different parts of a building are used during different hours of the day, they should have separate supply mains so that the heat can be entirely shut off in the unoccupied portion.

The orifice method of heat control and the intermittent method, both described in Chap. XI, are desirable in certain types of buildings. They require fairly intelligent operation to produce results.

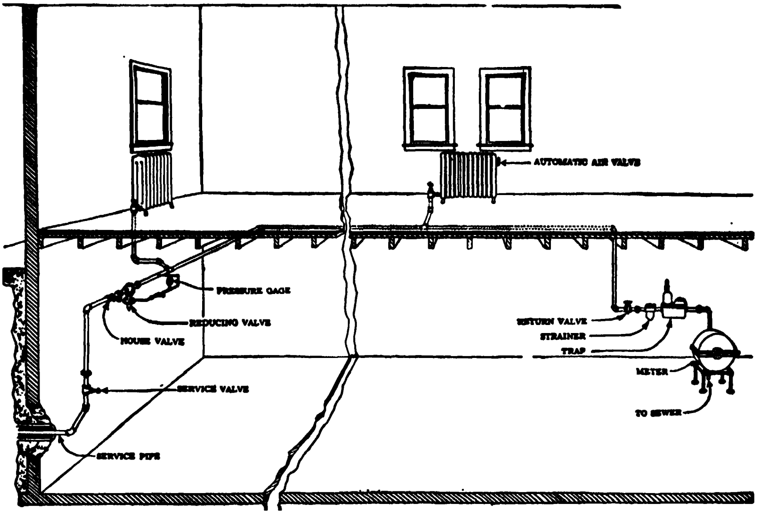

Fig. 189.—Consumer's piping system.

Another important provision is to remove the heat from the condensation before it is wasted to the sewer. The most satisfactory way to do this is to use it to preheat the water used in the lavatories and thus reduce the amount of steam used for water heating.

216. Meters.—The simplest method of measuring the amount of steam used is to measure the water of condensation as it is discharged from the return piping. The disadvantage is that any steam or condensate that leaks from the consumer's piping is not measured, but the simplicity of the meter as compared with other methods makes it widely used. One design of condensation meter is shown in Fig. 190. It consists of a drum revolved by the weight of the water, the revolutions being recorded on a dial.

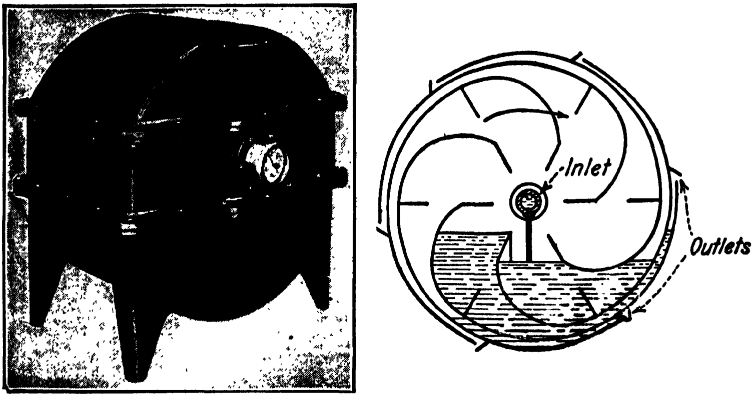

Fig. 190.—Condensation meter.

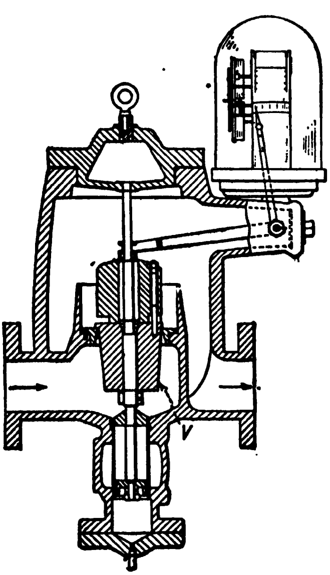

Fig. 191.—Steam-flow meter—displacement type.

It is absolutely necessary that the water flow freely away from the water outlet and that the incoming water have a reasonably steady flow and be not accompanied by steam, as may occur if a trap is leaky.

In some cases it is preferable to meter the steam at the point where it enters the consumer's piping system. The meter shown in Fig. 191 is of the displacement type, the plug *V* being raised by the steam flow to a position that is a function of the rate of flow. The chart records the quantity of steam that has passed through.

217. Electric Heating.—The use of electricity for building heating is ideal when it is economically practicable. It is successfully used in a few localities where the electricity is generated from water power and where circumstances make the current cost very low. Also in very mild climates, where heat is seldom needed, the convenience and cleanliness of its use warrant the higher cost. But in most parts of the United States electrical heating is seldom economically justified.

Tabulated below are typical costs for various fuels, which may be compared with electricity. It should be recognized that these fuel costs are extremely variable, depending on geographical location and quantity of fuel used.

Fuel	Assumptions			Cost per 1,000,000 B.t.u. delivered to the heating system
	Heating value, B.t.u.	Unit cost	Efficiency,[1] per cent	
Bituminous coal	14,000 per lb.	$10 per ton	65	$0.55
Coke	12,500 per lb.	$12 per ton	65	0.74
Anthracite coal	12,800 per lb.	$15 per ton	65	0.90
Oil	140,000 per gal.	$0.075 per gal.	70	0.77
Natural gas	1,000 per cu. ft.	$0.55 per 1,000 cu. ft.	75	0.74
Manufactured gas	550 per cu. ft.	$0.50 per 1,000 cu. ft.	75	1.21
Electricity	3,413 per kw.-hr.	½ cent per kw.-hr.	100	1.46
Electricity	3,413 per kw.-hr.	2½ cents per kw.-hr.	100	7.30

[1] Depends on type of heating system and whether hand fired or automatically fired by a conversion or combination unit.

The cost of electricity, like that of fuels, varies in different localities. Its range is from perhaps ½ to 2½ cents or more per kilowatt-hour.

Methods devised for using electricity during the so-called "off-peak" hours show possibilities of fairly reasonable costs. Some electric companies are willing to make lower rates for electricity used during hours when the total demand on their power stations is low. To take advantage of this it is necessary to provide means for storing the heat that is generated from the electricity (mostly during the night) and for utilizing it during the daytime.

This method is particularly applicable to hot-water systems. The water is heated by electricity to a temperature of 212° or slightly higher and is stored in tanks. When heat is required, the water is withdrawn and mixed, as needed, with the circulating water in the heating system.

The storage method may also be used in connection with a warm-air heating system. In such cases the stored hot water circulates through a tubular heater through which the air passes on its way to the rooms. The air pipes are arranged like those in an ordinary warm-air system.

The objections to the storage method are the cost of and the space occupied by the storage tanks. The cost of operation is still very much greater than that of oil or gas, but the future may witness some increase in the use of this method of heating. It can also be applied to the heating of the domestic water supply for lavatories, etc.

To overcome the objection of large storage facilities, a system[1] has been devised for storing water in a relatively small unfired vessel under a pressure of 200 lb. per sq. in. The tank normally operates 90 per cent full of water and the liquid is gradually converted to steam as the pressure drops in the tank due to the demand for heat. The vapor is released through a reducing valve that drops the pressure to 2 lb. per sq. in., which then may be used for circulation through a radiator system or a fin-type coil in a warm-air system.

Panel heating accomplished through various forms of electricity has already been referred to in Art. 82. Thus far the practical application of such heating has been confined principally to temporary or auxiliary purposes in the form of high-temperature heating elements that concentrate the heat rays in localized directions.

Another form of electric heating that may be developed further in the future is that heat derived from reversing the operating cycle of a refrigerating machine. A detailed explanation of this cycle will be found in Chap. XVIII.

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Problem

1. Figure 192 represents a central heating system for a group of institutional buildings. The figures opposite the arrowheads are the quantities of steam required by the various buildings, in pounds per hour. The lengths given are the pipe lengths between the points indicated by capital letters. The steam pressure at point A is 10 lb. per sq. in., and the minimum pressure to be supplied to any building is 2 lb. Calculate the pipe sizes.

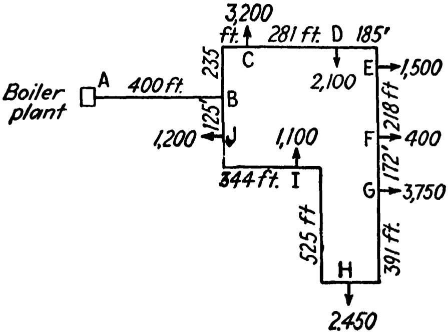

FIG. 192.

It is best to assume that half of the steam required by building G comes from each direction. It is not necessary to select pipe sizes to give exactly the total permissible pressure drop, and in a system of this sort it is undesirable to use pipes smaller than 4 in.

CHAPTER XIII

AIR AND ITS PROPERTIES

218. Composition of Air.—The atmosphere of the earth is a mixture of several gases, the proportions of which vary slightly in different localities. Water vapor is an important constituent, its amount being quite variable under different weather conditions.

The composition of air by volume, exclusive of water vapor, is given by Humphreys as follows:

	Per Cent
Dry air	100.00
Nitrogen	78.03
Oxygen	20.99
Argon	0.9323
Carbon dioxide	0.03
Hydrogen	0.01
Neon	0.0018
Krypton	0.0001
Helium	0.0005
Ozone	0.00006
Xenon	0.000009

Air is not a chemical combination but is a mechanical mixture of these gases.

Oxygen (O), which constitutes about one-fifth of the air by volume, is the element upon which animal life is dependent for its existence. In the process of respiration the lungs draw in and expel periodically a small quantity of air, and a portion of the oxygen unites chemically, while in the lungs, with impurities of the blood and thereby cleanses it. Some of the resulting products of this chemical reaction are exhaled in the form of gases and vapor. Human health and bodily comfort are dependent upon the proper performance of this process.

Nitrogen (N), which constitutes nearly all of the remaining four-fifths of the air by volume, is a relatively inert gas. It performs the function, essential for the human body, of diluting the oxygen; an atmosphere of pure oxygen would soon burn up and destroy the body tissues.

Carbon dioxide (CO_2) exists in small amounts in the open air, the purest air containing from 3 to 4 parts of CO_2 by volume in 10,000. It is also known as carbonic acid gas, as it forms a weak acid when dissolved in water. Being one of the products of respiration, it is found in larger quantities in the air of occupied rooms and was for a long time believed to have a poisonous effect when taken into the lungs, but is now known to be quite harmless, of itself, even in appreciable amounts. It has the effect, however, of diluting the oxygen content of the air, which necessitates an increase in the rate of breathing and under extreme conditions causes great discomfort. Haldane and Priestly found that with 2 per cent of CO_2 the lung action was increased 50 per cent; with 3 per cent of CO_2 about 100 per cent; with 4 per cent of CO_2 about 200 per cent; and with 6 per cent of CO_2 about 500 per cent. With 6 per cent, breathing becomes very difficult; and with more than 10 per cent there occurs a loss of consciousness but no immediate danger to life. Exposure to an atmosphere containing even 25 per cent of CO_2 does not result in immediate death.

As CO_2 is a product of respiration, the amount of it present in the atmosphere of a room is an indication of the amount of air being supplied, and the measurement of the CO_2 content of air is therefore of occasional significance in ventilating work.

Ozone (O_3) exists in minute quantities in the atmosphere, particularly in the country. Its amount varies considerably, and it is probably seldom present in quantities greater than 1 part in 10,000,000. Ozone is produced by electrical discharge and is generated artificially for use in ventilation. It has the property of partially destroying or masking the unpleasant odors that exist in buildings where there are large numbers of people. Its characteristic odor is somewhat like that of chlorine.

219. Ionization.—One property of air that is thought to be of possible importance in air conditioning is its ionic content. All gases contain atoms, molecules, or groups of molecules that carry positive or negative electrical charges and are called **ions.** In air there are two classes of ions, the small ions consisting of single molecules, and those found by the clustering of small ions about nucleuses such as droplets of condensation or dust particles. Large ions are more prevelant in city air. This classification as to size is not rigid, and ions of intermediate sizes are commonly found.

Ions are produced in nature by solar radiation, by cosmic rays, and by radioactive elements in the soil. They can be produced artificially by electrical means. The ion content of the atmosphere can be determined by measuring the aggregate of the electrical charges of the ions in a known volume of air.

220. Water Vapor.—Water vapor is an important constituent of the atmosphere. It is the most variable in quantity of any of the atmospheric elements because its amount depends largely on the weather conditions. In the northern part of the United States the range of the moisture content of the atmosphere is very great. In New York, for example, it varies at different times from ½ to 7 gr. per cu. ft. Water vapor, strictly speaking, is nothing other than *steam at very low pressures*, and its properties are those of steam. This fact should always be borne in mind.

Another conception that should be thoroughly understood is that of Dalton's law of partial pressures. Recent theories discussed in the Appendix indicate that this law is not scientifically correct because of the intermolecular forces taking place between dry air and water vapor, but for all practical purposes it is sufficiently accurate for air-conditioning calculations.

According to Dalton's law, **in any mechanical mixture of gases, each gas has a partial pressure of its own that is entirely independent of the partial pressures of the other gases.** For example, consider 1 cu. ft. of hydrogen gas at an absolute pressure of 5 lb. per sq. in. If 1 cu. ft. of nitrogen at an initial pressure of 10 lb. per sq. in. is injected into the same space, the resulting total pressure will be 15 lb. per sq. in., and the volume 1 cu. ft. In air, therefore, the oxygen, nitrogen, water vapor, and other gases each have their own partial pressure, the sum of all of them being equal to the total or barometric pressure.

For every temperature there is a corresponding partial pressure of water vapor at which the vapor is in a **saturated** state, its condition then being exactly similar to that of saturated steam. When the water vapor is in a saturated condition, the air is also spoken of as being saturated, since it then contains the maximum weight of vapor that it can hold at that temperature. If the temperature of the air is higher than that corresponding to the partial pressure of the water vapor, the vapor is **superheated;**

if the temperature drops below the saturation point, some of the vapor is condensed, and the vapor pressure is lowered to that corresponding to the new temperature. The saturation temperature is termed the **dew point.**

The partial pressure of saturated vapor increases as the temperature increases. Consequently air at higher temperatures is capable of holding a greater weight of water per cubic foot. It should be remembered that *the water vapor exists independently of the air except for the temperature effect of the latter;* and the vapor may be thought of as occupying the given volume at its own partial pressure. The intimate mixture of the air and vapor causes their temperatures to be the same.

221. Relative and Absolute Humidity.—Atmospheric moisture is termed **humidity. Absolute humidity** is the actual vapor content expressed in grains per cubic foot or per pound of air. The ratio of the vapor pressure to the vapor pressure of saturated air at the same temperature, expressed in per cent, is called the **relative humidity.** This is nearly the same as the ratio of the weight of moisture in the air to the weight of moisture in saturated air. For example, given a sample of air at 70° having an absolute humidity of 4 gr. per cu. ft. Since saturated air at 70° contains 8 gr. per cu. ft., the relative humidity is very nearly 50 per cent.

The capacity of air for absorbing moisture is due entirely to its temperature. When warm air is brought into contact with water, evaporation takes place, and the system attempts, so to speak, to reach that state of equilibrium in which the vapor pressure corresponds to the temperature.

222. Adiabatic Saturation.—When air below saturation is brought into intimate contact with water, there is always a tendency for some of the water to vaporize, adding to the moisture content of the air. If no heat is added from an outside source, the heat of vaporization for the added moisture will be supplied entirely at the expense of the heat of the air and of the superheat of the original quantity of water vapor. The evaporation of moisture and the lowering of the air temperature will continue until the air becomes saturated with water vapor.

A process of this nature taking place without a transfer of heat to or from an outside source is called **adiabatic,** and the final temperature that is reached is therefore termed the temperature of **adiabatic saturation,** which is almost identical with the wet-bulb temperature. This apparent identity is due to a set of favorable circumstances. Actually the mixing of water vapor and air taking place near the wetted surface of a wet-bulb thermometer is not adiabatic, because the bulb "sees" objects at the dry-bulb temperature. The result is a radiation error and a *higher* reading than true adiabatic saturation. Also, because of the difference in the rate of diffusion between air and water vapor, a wet-bulb thermometer will tend to read *lower* than the true temperature of adiabatic saturation. Hence these two errors largely compensate for each other and account for the close agreement between the observed wet-bulb temperature and the true temperature of adiabatic saturation, which Arnold[1] has shown to be a fortunate circumstance for the range of temperatures used in air-conditioning calculations.

For all practical purposes the wet-bulb temperature furnishes a means of determining the moisture content of air, inasmuch as its depression below the original temperature of the air will depend upon the amount of moisture that was added to bring the air to saturation. If the air is already saturated, no moisture can be added, and the wet-bulb and dry-bulb temperatures coincide.

The heat used in the vaporization of the added moisture is exactly equal to the heat given up by the air and by the water vapor originally contained, assuming that the added water was at the temperature of adiabatic saturation. This may be expressed mathematically as follows:[2]

Let

t = dry-bulb temperature of the air, deg. F.

t' = temperature of adiabatic saturation, or wet-bulb temperature, deg. F.

W' = weight of water vapor mixed with 1 lb. of dry air at saturation at temperature t', lb.

W = weight of water vapor mixed with 1 lb. of dry air at temperature t, lb.

$W' - W$ = weight of water added per pound of dry air, lb.

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[1] J. H. Arnold, "The Theory of the Psychrometer," *Physics*, Vol. 4, July and September, 1933.

[2] W. H. Carrier, "Rational Psychrometric Formulae," *Trans. A.S.M.E.*, 1911.

h_{fg} = latent heat of vaporization at temperature t', B.t.u. per lb.
C_{ps} = specific heat of water vapor at constant pressure.
C_{pa} = specific heat of dry air at constant pressure.

$$(W' - W)h_{fg} = C_{ps}W(t - t') + C_{pa}(t - t') \qquad (1)$$

or

$$W = \frac{h_{fg}W' - C_{pa}(t - t')}{h_{fg} + C_{ps}(t - t')} \qquad (2)$$

The distinction between the wet-bulb temperature and the dew point should be clearly understood. The wet-bulb temperature is produced by *adding moisture* to the air and causing its temperature to fall by reason of the giving up of heat to vaporize the water. The dew point, on the other hand, is reached by *removing heat* from the air without changing its moisture content.

The outdoor wet-bulb temperature in the United States rarely exceeds 80° in any region, and then only for short periods. The figure of 80° is the maximum wet-bulb temperature used as a design figure, and that only for certain Southern cities. For New York and Chicago the design wet-bulb temperature ordinarily used is 75° and the corresponding dry-bulb temperature is 95°.

223. Measurement of Wet-bulb Temperature.—In air-conditioning work the wet-bulb temperature is a basic factor, and its correct measurement is not only extremely important but also rather difficult. The stationary wet-bulb thermometer is of little value; a rapid movement of air over the wet bulb is essential. Carrier states[1] that the error in wet-bulb temperature measurements for different air velocities is approximately as follows:

Air Velocity, Ft. per Minute	Error in Wet-bulb Depression, Per Cent
0	15.0
500	4.5
1,000	2.7
2,000	1.5
4,000	0.8

True readings of wet-bulb temperature may be closely obtained if extremely high velocities of 5,000 ft. per min. or more are swept over the bulb. Even at these high velocities it is important to shield the thermometer unless the boundary surface temperatures of the test room or duct are nearly equal to the temperature conditions of the air under test.

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[1] W. H. CARRIER, "The Temperature of Evaporation," *Trans. A.S.H. & V.E.*, 1918.

The sling psychrometer illustrated in Fig. 3, Chap. I, is the most common instrument for the purpose and gives reasonably accurate results but is subject to certain errors, and the temperature indicated is slightly higher than the true wet-bulb temperature. The errors are due to (*a*) radiation from surrounding objects to the wet bulb, (*b*) conduction of heat along the stem of the thermometer, and (*c*) transmission of heat through the film of cooler air around the bulb.

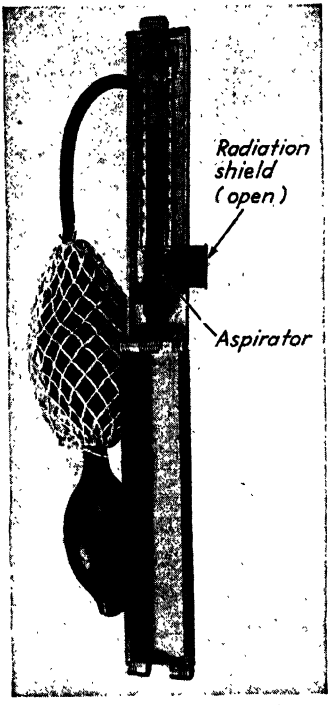

Fig. 193.—Hand-aspirated psychrometer. (*Julien P. Friez and Sons, Inc.*)

The psychrometer illustrated in Fig. 193 is designed to reduce these errors. Both thermometer bulbs are enclosed by a shield to prevent radiation, and the air is aspirated over the bulbs by means of a hand-operated rubber pumping bulb having a rubber pressure balloon.

224. Wet-bulb Temperature a Measure of Total Heat.—From a consideration of Eq. (1) and the process that it illustrates of adding water adiabatically, it is evident that the total heat of different mixtures of air and water vapor having the same wet-bulb temperature will be almost exactly equal. Thus if air is passed through a water spray, its moisture content will be increased and its dry-bulb temperature lowered, but its total heat and wet-bulb temperature will be unchanged, provided the spray water is at the wet-bulb temperature so that no heat is added or removed in the process. This is an important principle.

Furthermore, if the spray water having an initial temperature is continuously recirculated, the system will eventually reach a state of equilibrium in which the spray-water temperature will reach the wet-bulb temperature of the air. This process is essentially what takes place in the wet-bulb thermometer.

It should be remembered that the total heat and most other quantities in psychrometric work are based on the unit of 1 lb. of dry air, which is the quantity of a mixture that would weigh 1 lb. if all the water vapor were extracted. This is a convenient quantity, as the weight of *dry air* remains constant in processes involving humidification and dehumidification, while the volume and total weight of the mixture may both change.

The expression "1 lb. of dry air saturated with moisture" simply means 1 lb. of dry air plus the maximum amount of water vapor that can be associated with that quantity of air at that temperature.

225. Vapor Pressure.—In a mixture of air and water vapor the total or barometric pressure is the sum of the partial pressures of the gases constituting the air, and the partial pressure of the water vapor. Theoretically this assumption is not scientifically correct, as mentioned in Art. 220 and as discussed in the Appendix, but it is accurate enough for all practical purposes. Hence, for a given weight of water vapor per pound of dry air the respective partial pressures are fixed by the inherent properties of the gases. These are defined by the familiar laws of Boyle and Charles and expressed by the equation

$$PV = WRT \qquad (3)$$

where P = absolute pressure of the gas.
V = its volume.
W = its weight.
T = its absolute temperature ($T = t + 459.7$).
R = a constant for the particular gas.

When P is in pounds per square foot, V in cubic feet, W in pounds, and T in degs. F., the value for R for air averages 53.35, and for water vapor 85.78.[1]

In a mixture of air and water vapor that occupy the same volume and are at the same temperature

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[1] Calculated from international atomic weights.

$$\frac{P}{P'} = \frac{W \times 53.35}{W' \times 85.78} \tag{4}$$

where P and P' are the pressures and W and W' the weights of the air and water vapor, respectively.

Example 1.—Calculate the weight and specific volume of dry air at a temperature of 70° when the barometric pressure is 29.92 in. of mercury and compare it with the density and volume of 1 lb. of dry air saturated with moisture at the same temperature and barometric pressure.

(1 in. of mercury = 0.492 lb. per sq. in.)

Utilizing Eq. (3), the weight of dry air is

$$W = \frac{144 \times 0.492 \times 29.92 \times 1}{53.35(459.7 + 70)} = 0.07492 \text{ lb. per cu. ft.}$$

The weight of the saturated air is determined by calculating separately the density of the dry air and the vapor as follows:

From Table 65*, the pressure corresponding to 70° steam or water vapor is 0.7392 in. of mercury, and the weight of vapor is

$$W = \frac{144 \times 0.492 \times 0.7392 \times 1}{85.78(459.7 + 70)} = 0.001152 \text{ lb. per cu. ft.}$$

The pressure exerted by the dry air in the saturated mixture is

$$29.92 - 0.7392 = 29.1808 \text{ in. of mercury,}$$

and the weight of dry air is

$$W = \frac{144 \times 0.492 \times 29.1808 \times 1}{53.35(459.7 + 70)} = 0.07308 \text{ lb. per cu. ft.}$$

Total weight of saturated air is 0.001152 plus 0.07308, or 0.074232 lb. per cu. ft.

The specific volume for dry air is the reciprocal of the density, or

$$1 \div 0.07492 = 13.34 \text{ cu. ft. per lb.}$$

The *volume* of saturated air *per pound of dry air* is a useful quantity in air-conditioning calculations, but should not be confused with *specific* volume, which refers to volume per pound of mixture. The *volume* may be determined by dividing the *weight* of saturated air per pound of dry air by the *weight* or density of saturated air per unit volume. For the example above the weight of saturated air is, of course, 1 lb. of dry air plus 0.01574 (weight of 70° saturated vapor, from Table 66). Then the volume is 1.01574 ÷ 0.074232 = 13.68 cu. ft. per lb. of dry air, which is the value listed in Table 66 and is also shown on the pychrometric chart.

It will be noted that all the above results check closely with the corresponding values in Tables 63, 64, and 66. In fact, the computation procedure outlined in this example is the method customarily followed in developing these data. Small discrepancies are principally due to the different values of vapor pressure, which will vary slightly depending on the source of the data.

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* The tables accompanying this chapter are at the end of the chapter.

In the symbols commonly used in air conditioning, taking 1 lb. of dry air (with its accompanying water vapor)

$$\frac{B - e}{e} = \frac{7{,}000}{w} \times \frac{53.35}{85.78} \tag{5}$$

where B = the barometric pressure, in. of Hg.
e = pressure of the water vapor, in. of Hg.
w = weight of water vapor, grains.
7,000 = grains per lb.

Or

$$e = \frac{Bw}{4{,}353 + w} \tag{6}$$

Or

$$w = \frac{4353e}{B - e} \tag{7}$$

Thus if the weight of water vapor per pound of dry air is known, the vapor pressure can be determined, or vice versa.

Example 2.—Compute weight of saturated vapor per pound of dry air for 70° air if the barometric pressure is 29.92 in. of mercury. From Table 65, the pressure of the water vapor corresponding to 70° is $e = 0.7392$ in. of mercury.

Using Eq. (7), the weight of water vapor is

$$w = \frac{4353 \times 0.7392}{29.92 - 0.7392} = 110.2 \text{ gr. per lb. of dry air}$$

or

$110.2 \div 7{,}000 = 0.01574$ lb. per lb. of dry air.

Both of these values check with the data in Table 66.

The vapor pressure and consequently the weight of water vapor can be determined from the wet- and dry-bulb temperatures by means of Carrier's equation

$$e = e' - \frac{(B - e')(t - t')}{2{,}800 - 1.3t'} \tag{8}$$

where e' = saturation pressure at the wet-bulb temperature (from the steam, Table 65).

t = dry-bulb temperature, deg. F.
t' = wet-bulb temperature, deg. F.
and the other symbols are as in Eq. (5).

Example 3.—Given air with a dry-bulb temperature of 80°, a wet-bulb temperature of 67°, and a barometric pressure of 24.80 in. of mercury (corresponds to an altitude of about 5,000 ft.). Calculate the relative humidity and density of 1 cu. ft. of the mixture.

From Table 65, the saturation pressure at 67° wet-bulb temperature is $e' = 0.6669$. Substituting in Eq. (8), the pressure of the water vapor in the mixture is

$$e = 0.6669 - \frac{(24.80 - 0.6669)(80 - 67)}{2{,}800 - (1.3 \times 67)} = 0.5508 \text{ in. of mercury}$$

Relative humidity (Φ) by definition is the ratio of the actual water vapor pressure (e) to the vapor pressure of saturated air (e_s) at the same temperature. Thus, from Table 65, the saturated vapor pressure of 80° dry-bulb temperature is $e_s = 1.0321$, and

$$\Phi = \frac{e}{e_s} = \frac{0.5508}{1.0321} \times 100 = 53.3 \text{ per cent}$$

From Eq. (3), the weight of vapor in the mixture is

$$W = \frac{144 \times 0.492 \times 0.5508 \times 1}{85.78(459.7 + 80)} = \frac{0.8261 \times 0.5508}{539.7}$$
$$W = 0.000843 \text{ lb. per cu. ft.}$$

Pressure of the dry air in the mixture is 24.80 − 0.5508 = 24.2492 in. of mercury, and the weight of dry air is

$$W = \frac{144 \times 0.492 \times 24.2492 \times 1}{53.35(459.7 + 80)} = \frac{1.328 \times 24.2492}{539.7}$$
$$W = 0.05967 \text{ lb. per cu. ft.}$$

The total weight or density of the mixture is

0.000843 plus 0.05967 = 0.060513 lb. per cu. ft.

225. Specific Heat of Air and Water Vapor.—The specific heat of air varies slightly with temperature. The values given in the International Critical Tables (converted to English units) for dry air are

$$\text{At } 32°\ C_{pa} = 0.2399$$
$$\text{At } 212°\ C_{pa} = 0.2403$$

For air-conditioning work the figure 0.24 is appropriate. The **sensible heat** of pure dry air is therefore

$$H_s = C_{pa}(t_2 - t_1) = 0.24(t_2 - t_1) \qquad (9)$$

where H_s = sensible heat above t_1, B.t.u. per lb.
t_2 = dry-bulb air temperature, deg. F.
t_1 = base temperature above which the sensible heat is to be determined, deg. F.

The specific heat of water vapor (C_{ps})has been found to vary in accordance with the following relationship:

$$C_{ps} = 0.4423 + 0.00018t \qquad (10)$$

where t is the temperature in deg. F.

At 70°, C_{ps} thus becomes 0.4549.

As heat must be added to change water into steam, likewise it requires heat to evaporate water at temperatures below the atmospheric boiling point. The heat required to perform this change is termed **latent heat** and its amount will vary with temperature. For temperatures between 40 and 150° the latent heat of water vapor may be derived from Eq. (11), or it may be obtained from Table 65.

$$h_{fg} = 1094.5 - 0.575t \qquad (11)$$

where h_{fg} = latent heat of water vapor, B.t.u. per lb.
t = temperature, deg. F.

227. Total Heat, or Enthalpy.—The total heat of air containing aqueous vapor is the sum of the *heat* in the air and the *heat* in the vapor. If it were necessary to know the *absolute* total heat, then it would be necessary to include all heat above the absolute zero; but for air-conditioning work it is more practicable to choose another datum above which the "total" heat is calculated. Misunderstanding as to what this datum is has caused confusion, and it is desirable that the student keep it clearly in mind.

The total heat, as originally defined by Carrier and hereafter referred to as the **Carrier total heat,** consists of (*a*) the heat in the air above zero and (*b*) the heat of vaporization of the water vapor plus its superheat. Thus the heat of the liquid in the water vapor is not included.

Taking Eq. (1) (Art. 222) and rearranging it

$$C_{pa}t' + W'h_{fg} = C_{pa}t + W[h_{fg} + C_{ps}(t - t')] \qquad (12)$$

The total heat at t' is obviously represented by the left-hand side of the equation; and since the total heat at t is assumed to be equal to the total heat at t' (Art. 222), then

$$H = C_{pa}t + W[h_{fg} + C_{ps}(t - t')] \qquad (13)$$

This equation may be simplified to the form

$$H = 0.24t + Wh_{fg} + 0.45W(t - t') \qquad (14)$$

Example 1.—Find the Carrier total heat of a mixture of dry air and water vapor having a dry-bulb temperature of 80° and a wet-bulb temperature of 67° if the barometric pressure is 29.92 in. of mercury.

From Table 65, $e^1 = 0.6669$ in. of mercury, and then according to Eq. (8), the pressure of the water vapor in the mixture is

$$e = 0.6669 - \frac{(29.92 - 0.6669)(80 - 67)}{2{,}800 - (1.3 \times 67)}$$
$$e = 0.6669 - 0.1401 = 0.5268 \text{ in. of Hg}$$

Using Eq. (7), the weight of water vapor in the mixture is

$$w = \frac{4353 \times 0.5268}{29.92 - 0.5268} = 78.01 \text{ gr.} \div 7{,}000 = 0.01114 \text{ lb. per lb. dry air}$$

From Table 65, h_{fg} corresponding to the wet-bulb temperature = 1056.0 B.t.u. per lb. The Carrier total heat may then be calculated from Eq. (14) as follows:

$$H = (0.24 \times 80) + (0.01114 \times 1056) + 0.45 \times 0.01114(80 - 67)$$
$$H = 19.2 + 11.764 + 0.065 = 31.029 \text{ B.t.u. per lb. dry air}$$

For air at saturation Eq. (14) becomes

$$H = 0.24t' + Wh_{fg} \qquad (15)$$

Example 2.—Find the Carrier total heat of 1 lb. of 70° saturated air at a barometric pressure of 29.92 in. of mercury. From Example 2 in Art. 225, $w = 0.01574$ lb. per lb. dry air. Using Eqs. (11) and (15)

$$H = 0.24(70) + 0.01574(1094.5 - 0.575 \times 70)$$
$$H = 16.8 + 16.59 = 33.39 \text{ B.t.u. per lb. dry air}$$

This value does not check with the enthalpy in Table 66, as these data include the heat of the liquid, but the result obtained in this example does check with the value obtained from the psychrometric chart shown in Fig. 194.

Since the heat of the liquid in the water vapor is not included in the preceding definition of total heat, Eq. (12) becomes slightly incorrect when used in some problems involving condensation or evaporation of moisture. A more correct measure of heat content, termed **enthalpy**, uses 0° for the datum for dry air, and

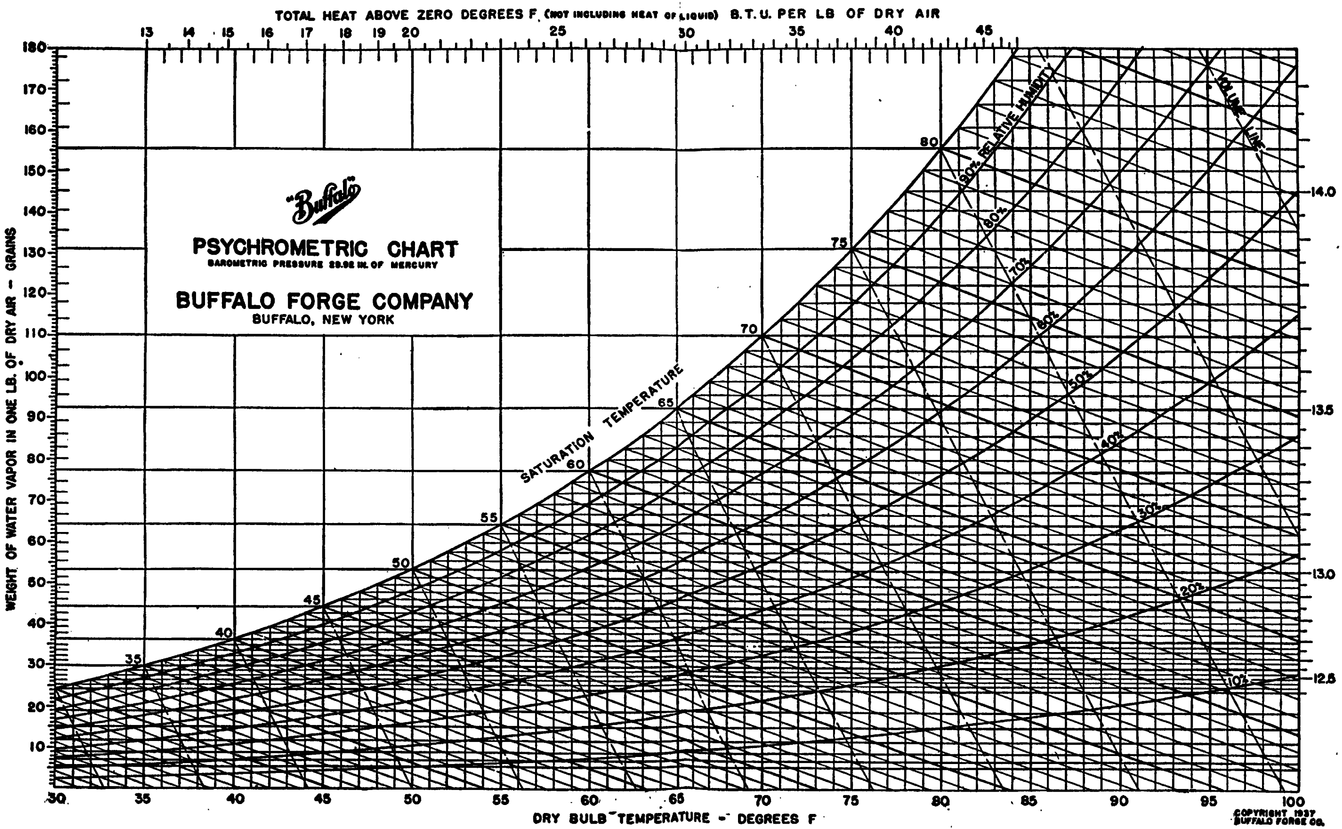

FIG. 194.—Psychrometric chart (*From "Fan Engineering," reproduced by permission of the Buffalo Forge Company.*)

liquid at 32° as the datum for water vapor. On this basis, Eq. (13) is modified as follows:

$$E = C_{pa}t + W[h_f + h_{fg} + C'_{ps}(t - t')] \tag{16}$$

where E = enthalpy, B.t.u. per lb. of dry air.

h_f = heat of the liquid above 32°, B.t.u. per lb.

and the other symbols correspond to those in Eq. (13). Note that the symbols and their numerical values are those obtained from Table 65.

Example 3.—Determine the enthalpy of 1 lb. of 70° saturated air at a barometric pressure of 29.92 in. of mercury. The value $w = 0.01574$ lb. per lb. dry air has already been determined. From Table 65, $h_f = 38.04$ and $h_{fg} = 1054.3$. From Eq. (16)

$$E = 0.24(70) + 0.01574[38.04 + 1054.3 + 0.45(70 - 70)]$$
$$E = 16.8 + 0.01574(1092.3)$$
$$E = 16.8 + 17.19 = 33.99 \text{ B.t.u. per lb. dry air}$$

This value checks closely with the enthalpy figure of Table 66. The discrepancy is due to the fact that vapor pressures used in compiling Table 66 are not in strict accord with those of Table 65. Furthermore, the column headed Vapor under Enthalpy in Table 66 was computed from an empirical relation $h_g = 1059.2 + C_{ps}\,t$. Substituting the proper values for the above example,

$$h_g = 1059.2 + 0.45(70) = 1090.7$$

Using this value in the above example

$$E = 16.8 + 0.01574(1090.7)$$
$$E = 16.8 + 17.16 = 33.96 \text{ B.t.u. per lb. dry air}$$

The objection to the use of enthalpy values in air-conditioning calculations is that the enthalpy does not remain uniform for constant temperature of adiabatic saturation (wet-bulb temperature) but varies with the dry-bulb temperature. Therefore it is simpler to use the Carrier total heat (sometimes referred to as Sigma function),[1] which does not include the heat of the liquid. This permits air and water-vapor calculations to be computed with standard psychrometric charts, and the necessary corrections can be made for the addition or extraction of water, depending on the temperature and weight of water under consideration.

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[1] Carrier and Mackey, "A Review of Existing Psychrometric Data in Relation to Practical Engineering Problems," *Trans. A.S.M.E.*, January, 1937.

A student interested in the exact physics of air and water-vapor mixtures will profit by consulting the references given at the end of this chapter, and also a section relating to this subject in the Appendix. In the engineering of air-conditioning systems, sufficiently accurate calculations can be made from psychrometric charts and tables.

228. Psychrometric Chart and Tables.—The psychrometric chart[1] (Fig. 194) shows graphically the properties of air on the basis of 1 lb. of dry air.[2] The chart in Fig. 3 (Chap. I) is on the basis of 1 cu. ft. of air. The diagram in Fig. 195 illustrates the method of using the chart. The properties of dry and saturated air are given, respectively, in Tables 63, 64, and 66.

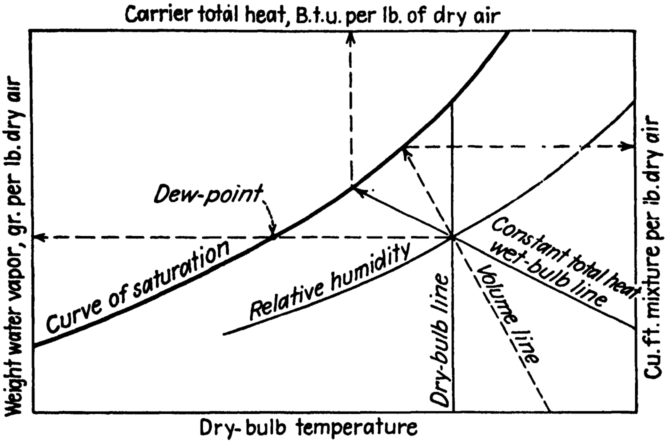
Carrier total heat, B.t.u. per lb. of dry air
Weight water vapor, gr. per lb. dry air
Cu. ft. mixture per lb. dry air
Dew-point
Curve of saturation
Relative humidity
Dry-bulb line
Volume line
Constant total heat wet-bulb line
Dry-bulb temperature

FIG. 195.—Method of using chart.

Example.—From a suitable psychrometer it is determined that air has a dry-bulb temperature of 80° and a wet-bulb temperature of 67° with a barometric pressure of 29.92 in. of mercury. From the psychrometric chart find relative humidity, dew-point temperature, weight of water vapor in the mixture, Carrier total heat, and volume of the mixture.

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[1] From "Fan Engineering," reproduced by permission of the Buffalo Forge Company.

[2] The properties of air are commonly expressed as "per pound of dry air." Terms such as "B.t.u. per pound of dry air" or "grains of moisture per pound of dry air" mean, respectively, "B.t.u. content of a mixture of 1 lb. of dry air and its accompanying water vapor" and "grains of moisture accompanying 1 lb. of dry air."

Referring to the diagram in Fig. 195 and selecting proper values from Fig. 194, it will be found that at the intersection of the vertical 80° dry-bulb line and diagonal 67° wet-bulb line the relative humidity is 51 per cent. Moving horizontally from this intersection to the curve of saturation gives a 60.4° dew-point temperature. To obtain grains of water vapor per pound of dry air continue to left from the dew point to the scale on the left of the chart and read 78 gr., which checks with the result obtained by calculation in Example 1, Art. 227.

The Carrier total heat is found by following the 67° wet-bulb line to the curve of saturation and then vertically upward to the scale at the top of the chart, which reads 31.1 B.t.u. per lb. of dry air and also checks with the result in Example 1, Art. 227.

To obtain cubic feet of the moisture per pound of dry air, proceed from the intersection of the 80° dry-bulb and 67° wet-bulb temperature upward parallel with the inclined volume line (shown dotted) to the curve of saturation, and then directly to the right of the chart, where a value of 13.84 cu. ft. may be read.

A psychrometric chart for high temperatures is given in the Appendix.

There are numerous other psychrometric charts in existence, some based on the Carrier formula and others based on slightly different data and using enthalpy rather than total heat as defined in Art. 227.

229. Barometric Pressure Corrections.—All the tables at the end of this chapter and the psychrometric chart are based on a standard barometric pressure of 29.92 in. of mercury. No corrections are necessary for small barometric differences, but at higher altitudes it may be advisable in some problems to correct the readings from a standard chart or table. Absolutely accurate corrections are relatively complex, but generally great refinement is not necessary because of the local variations in barometric pressure.

There are only two properties of air that cannot be determined directly from a psychrometric chart for a barometric pressure other than standard. These are *moisture content* of the mixture and *relative humidity*, both of which may be calculated according to procedures previously referred to. The following example will illustrate the method of making the necessary corrections.

Example 1.—Suppose that the dry-bulb and wet-bulb temperatures were found to be 80° and 67°, respectively. The local mean barometric pressure is 24.80 in. of mercury which corresponds to an altitude of about 5,000 ft. It is desired to determine the properties of the air.

Moisture content may be determined according to Eqs. (7) and (8). As calculated from Eq. (8) in Example 3, Art. 225, the pressure of water vapor in the mixture is 0.5508 in. of mercury. Substituting this value in Eq. (7) gives the weight of water vapor

$$w = \frac{4353 \times 0.5508}{24.3492} = 98.4 \text{ gr. per lb. dry air}$$

Relative humidity is determined by first calculating the pressure of the water vapor according to Eq. (8) and then dividing this value by the pressure of saturated vapor at 80°. According to Example 3, Art. 225, the relative humidity is 53.3 per cent.

Dew-point temperature may be established by reading from Table 65 the temperature corresponding to the pressure of the water vapor = 0.5508 in. of mercury as calculated from Eq. (8). This temperature is 61.5°.

Carrier total heat may be determined directly from the standard psychrometric chart of Fig. 194 by locating a point at the intersection of the 80° dry-bulb line and the moisture content of 98.4 gr. per lb. This point lies on the 70.8° wet-bulb line and the Carrier total heat is 34.2 B.t.u. per lb. dry air.

Specific volume as read from the standard chart is 13.84 cu. ft. Volume at a given dry-bulb temperature and moisture content is inversely proportional to the barometric pressure.

$$V = 13.84 \times \frac{29.92}{24.80} = 16.7 \text{ cu. ft. per lb.}$$

The following tabulation summarizes the above corrections for a barometric pressure of 24.80 in. of mercury, and in addition values from the Example in Art. 228 are included corresponding to standard barometric pressure of 29.92 in. of mercury.

Barometric Pressure, In. of Hg	24.80	29.92
Dry-bulb temperature, °F	80.0	80.0
Wet-bulb temperature, °F	67.0	67.0
Dew-point temperature, °F	61.5	60.4
Relative humidity, per cent	53.3	51.0
Moisture content, gr. per lb. dry air	98.4	78.0
Carrier total heat, B.t.u. per lb. dry air	34.2	31.1
Volume, cu. ft. per lb	16.7	13.84

Example 2.—In fan testing, standard air conditions for rating purposes are considered to be 70° dry-bulb temperature and 29.92 in. of mercury barometric pressure. Suppose that the fan connected to a warm-air furnace heats the air to a temperature of 140° as measured at the outlet of the unit, where an air-measuring instrument indicates that 1,600 cu. ft. of air per min. is being circulated. The actual barometric pressure at the time of test is 29.1 in. of mercury. Convert these test conditions to standard air.

The volume of a given weight of air varies directly as the absolute temperature and inversely as the barometric pressure. Let Q be the volume for temperature t and barometer b and Q_s the corresponding volume for temperature t_s and barometer b_s. Then

$$Q_s = Q\left(\frac{459.7 + t_s}{459.7 + t}\right)\left(\frac{b}{b_s}\right)$$

$$Q_s = 1{,}600\left(\frac{459.7 + 70}{459.7 + 140}\right)\left(\frac{29.10}{29.92}\right)$$

$$Q_s = 1{,}600 \times 0.883 \times 0.972 = 1373 \text{ c.f.m. at standard air conditions}$$

230. Mixtures.—Problems involving mixtures of air of different temperatures and moisture contents are often encountered and can be solved by means of the psychrometric chart.

Example.—One hundred fifty pounds of air at a dry-bulb temperature of 60° and a wet-bulb temperature of 55° is mixed with 350 lb. of air at a dry-bulb temperature of 80° and a wet-bulb temperature of 65°. What are the resulting dry- and wet-bulb temperatures?

Total heat per pound for the 60° air is 23.0 B.t.u.; of the 80° air, 29.6 B.t.u.

$$\begin{aligned} 150 \text{ lb.} \times 23.0 &= 3{,}450 \text{ B.t.u.} \\ 350 \text{ lb.} \times 29.6 &= \underline{10{,}360} \text{ B.t.u.} \\ \text{Total quantity of heat} &= 13{,}810 \text{ B.t.u.} \end{aligned}$$

$$\text{Heat per pound of dry air in mixture} = \frac{13{,}810}{500} = 27.6 \text{ B.t.u.}$$

Weight of moisture per pound for the 60° air is 56.0 gr.; for the 80° air, 68.0 gr.

$$\begin{aligned} 150 \text{ lb.} \times 56.0 &= 8{,}400 \text{ gr.} \\ 350 \text{ lb.} \times 68.0 &= \underline{23{,}800} \text{ gr.} \\ \text{Total quantity of moisture} &= 32{,}200 \text{ gr.} \end{aligned}$$

$$\text{Moisture per pound of dry air in mixture} = \frac{32{,}200}{500} = 64.4 \text{ gr.}$$

The value 27.6 B.t.u. per lb. corresponds to a wet-bulb temperature of 62.2°. The dry-bulb temperature corresponding to a wet-bulb of 62.2° and a moisture content of 64.4 gr. is 74.2°. The mixture would therefore have a dry-bulb temperature of 74.2°, a wet-bulb of 62.2, and a moisture content of 64.4 grains per pound.

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Problems

1. Given wet-bulb temperature 69°, dry-bulb temperature 81°. Find dew point, percentage of saturation, and moisture content.

2. Given air at a temperature of 60° and containing 55 gr. of moisture per pound of dry air. What is its relative humidity?

3. The air outside is at a temperature of 45° and has a relative humidity of 81 per cent. On being drawn into the building it is heated to 70°. What is its relative humidity at the higher temperature?

4. Air at 85° is 79 per cent saturated. When cooled to 65°, what is its new moisture content?

5. A stream of air is sprayed with water, and moisture is thus added adiabatically. The original condition is dry bulb 85°, wet bulb 70°. The final dry-bulb temperature is 75°. How much moisture was added per pound of dry air?

6. Air at 85° and containing 130 gr. per lb. of dry air is cooled to 60° and reheated to its original temperature. What is its new wet-bulb temperature, and how much moisture was removed?

7. If 397 lb. of air at a dry-bulb temperature of 55° and a wet-bulb temperature of 50° is mixed with 900 lb. of air at a dry-bulb temperature of 82° and a wet-bulb temperature of 67°, what will be the dry- and wet-bulb temperatures of the mixture?

8. One pound of air at 90° dry-bulb temperature and 60 per cent humidity is cooled to 65°. The condensed moisture is removed. The same amount of heat as was removed is replaced. What will be the new wet-bulb and dry-bulb temperatures?

9. What would be the dry-bulb temperature, wet-bulb temperature, dew point, and relative humidity if the following quantities of air were mixed: 5,000 cu. ft. at 90° and 50 per cent humidity; 3,500 cu. ft. at 75° dry-bulb and 67° wet-bulb temperature; and 3,000 cu. ft. at 50° dry-bulb and 50° wet-bulb temperature?

10. It is desired to reduce the relative humidity of 950 lb. of air at 83° dry bulb from 63 to 35 per cent by cooling and reheating. How much heat must be removed, and how much added?

11. Given air at 73° dry bulb and 59° wet bulb, what are the vapor pressure and the weight of moisture per pound of dry air? Solve by calculation, not from the chart.

12. What is the enthalpy and what is the total heat of air at 75° dry bulb and 57° wet bulb? Determine by calculation.

13. A building of 42,100 cu. ft. of net internal volume is ventilated at the rate of three air renewals per hour. When the outside temperature is 0° and the humidity 80 per cent, how much heat and how much water must be added to the entering air if it is to be introduced at 70° and 30 per cent humidity?

14. Air at 95° dry bulb and 73° wet bulb is passed over a drying agent that reduces its dew point to 42° but releases the heat of vaporization of the moisture that has been removed, plus 160 B.t.u. per lb. of moisture removed which is due to a chemical reaction. The air is then passed through a spray chamber in which the spray water is recirculated and in which the air is brought to 50 per cent relative humidity. What are the final wet- and dry-bulb temperatures?

TABLE 63.—PROPERTIES OF DRY AIR[1]
(Barometric pressure 29.921 in. of Hg)

Temperature, °F.	Density, lb. per cu. ft.	Ratio vol. to vol. at 70°F.	B.t.u. absorbed by 1 cu. ft. dry air per °F.	Cu. ft. dry air warmed 1° per B.t.u.
0	0.08633	0.8678	0.02077	48.15
10	0.08449	0.8867	0.02030	49.26
20	0.08273	0.9056	0.01986	50.35
30	0.08104	0.9245	0.01944	51.44
40	0.07942	0.9433	0.01905	52.49
50	0.07785	0.9624	0.01868	53.36
60	0.07636	0.9811	0.01832	54.44
70	0.07492	1.0000	0.01798	55.62
80	0.07353	1.0189	0.01765	56.66
90	0.07219	1.0378	0.01733	57.70
100	0.07090	1.0567	0.01702	58.75
110	0.06966	1.0755	0.01672	59.81
120	0.06845	1.0946	0.01643	60.86
130	0.06729	1.1133	0.01616	61.88
140	0.06617	1.1322	0.01589	62.93
150	0.06509	1.1510	0.01563	63.98
160	0.06403	1.1701	0.01538	65.02
180	0.06203	1.2078	0.01490	67.11
200	0.06015	1.2456	0.01446	69.24

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[1] Compiled by W. H. Severns. Copyright, A.S.H. & V.E., from "Heating-Ventilating-Air Conditioning Guide," 1940, Chap. I.

TABLE 64.—PROPERTIES OF SATURATED AIR[1]

(Weights of air, vapor, and saturated mixture of air and vapor at 29.921 in. of Hg)

Temp., °F	Weight in cu. ft. of mixture			B.t.u. absorbed by 1 cu. ft. sat. air per °F.	Cu. ft. sat. air warmed 1° per B.t.u.	Specific heat, B.t.u. per lb. of mixture
	Weight of dry air, lb.	Weight of vapor, lb.	Total weight of mixture, lb.			
0	0.08622	0.000068	0.08629	0.02078	48.12	0.2408
10	0.08431	0.000111	0.08442	0.02031	49.24	0.2406
20	0.08244	0.000177	0.08262	0.01987	50.33	0.2405
30	0.08060	0.000278	0.08088	0.01946	51.39	0.2406
40	0.07876	0.000409	0.07917	0.01908	52.41	0.2410
50	0.07692	0.000587	0.07751	0.01872	53.42	0.2415
60	0.07503	0.000828	0.07586	0.01838	54.41	0.2423
70	0.07307	0.001151	0.07422	0.01805	55.40	0.2432
80	0.07099	0.001578	0.07257	0.01775	56.34	0.2446
90	0.06877	0.002134	0.07090	0.01747	57.24	0.2464
100	0.06634	0.002851	0.06919	0.01721	58.11	0.2487
110	0.06361	0.003762	0.06737	0.01696	58.96	0.2517
120	0.06057	0.004912	0.06548	0.01675	59.70	0.2558
130	0.05712	0.006344	0.06346	0.01657	60.35	0.2611
140	0.05317	0.008116	0.06129	0.01642	60.91	0.2679
150	0.04863	0.010284	0.05891	0.01630	61.35	0.2767
160	0.04339	0.012919	0.05631	0.01624	61.58	0.2884
170	0.03733	0.016092	0.05342	0.01621	61.69	0.3034
180	0.03033	0.019888	0.05022	0.01624	61.58	0.3234
190	0.02228	0.024384	0.04666	0.01633	61.24	0.3500
200	0.01298	0.029700	0.04268	0.01649	60.64	0.3864

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[1] Compiled by W. H. Severns. Copyright A.S.H. & V.E., from "Heating-Ventilating-Air Conditioning Guide," 1940, Chap. I.

TABLE 65.—PROPERTIES OF STEAM AT LOW PRESSURES[1]

Temp. °F.	Abs. pressure		Specific volume, cu. ft. per lb.			Total heat or enthalpy, B.t.u. per lb.		
	Lb. per sq. in.	In. Hg	Sat. liquid	Evap.	Sat. vapor	Sat. liquid	Evap.	Sat. vapor
t	p		v_f	v_{fg}	v_g	h_f	h_{fg}	h_g
32	0.08854	0.1803	0.01602	3306	3306	0.00	1075.8	1075.8
33	0.09223	0.1878	0.01602	3180	3180	1.01	1075.2	1076.2
34	0.09603	0.1955	0.01602	3061	3061	2.02	1074.7	1076.7
35	0.09995	0.2035	0.01602	2947	2947	3.02	1074.1	1077.1
36	0.10401	0.2118	0.01602	2837	2837	4.03	1073.6	1077.6
37	0.10821	0.2203	0.01602	2732	2732	5.04	1073.0	1078.0
38	0.11256	0.2292	0.01602	2632	2632	6.04	1072.4	1078.4
39	0.11705	0.2383	0.01602	2536	2536	7.04	1071.9	1078.9
40	0.12170	0.2478	0.01602	2444	2444	8.05	1071.3	1079.3
41	0.12652	0.2576	0.01602	2356	2356	9.05	1070.7	1079.7
42	0.13150	0.2677	0.01602	2271	2271	10.05	1070.1	1080.2
43	0.13665	0.2782	0.01602	2190	2190	11.06	1069.5	1080.6
44	0.14199	0.2891	0.01602	2112	2112	12.06	1068.9	1081.0
45	0.14752	0.3004	0.01602	2036.4	2036.4	13.06	1068.4	1081.5
46	0.15323	0.3120	0.01602	1964.3	1964.3	14.06	1067.8	1081.9
47	0.15914	0.3240	0.01603	1895.1	1895.1	15.07	1067.3	1082.4
48	0.16525	0.3364	0.01603	1828.6	1828.6	16.07	1066.7	1082.8
49	0.17157	0.3493	0.01603	1764.7	1764.7	17.07	1066.1	1083.2
50	0.17811	0.3626	0.01603	1703.2	1703.2	18.07	1065.6	1083.7
51	0.18486	0.3764	0.01603	1644.2	1644.2	19.07	1065.0	1084.1
52	0.19182	0.3906	0.01603	1587.6	1587.6	20.07	1064.4	1084.5
53	0.19900	0.4052	0.01603	1533.3	1533.3	21.07	1063.9	1085.0
54	0.20642	0.4203	0.01603	1481.0	1481.0	22.07	1063.3	1085.4
55	0.2141	0.4359	0.01603	1430.7	1430.7	23.07	1062.7	1085.8
56	0.2220	0.4520	0.01603	1382.4	1382.4	24.06	1062.2	1086.3
57	0.2302	0.4686	0.01603	1335.9	1335.9	25.06	1061.6	1086.7
58	0.2386	0.4858	0.01604	1291.1	1291.1	26.06	1061.0	1087.1
59	0.2473	0.5035	0.01604	1248.1	1248.1	27.06	1060.5	1087.6
60	0.2563	0.5218	0.01604	1206.6	1206.7	28.06	1059.6	1088.0
61	0.2655	0.5407	0.01604	1166.8	1166.8	29.06	1059.3	1088.4
62	0.2751	0.5601	0.01604	1128.4	1128.4	30.05	1058.8	1088.9
63	0.2850	0.5802	0.01604	1091.4	1091.4	31.05	1058.2	1089.3
64	0.2951	0.6009	0.01605	1055.7	1055.7	32.05	1057.6	1089.7
65	0.3056	0.6222	0.01605	1021.4	1021.4	33.05	1057.1	1090.2
66	0.3164	0.6442	0.01605	988.4	988.4	34.05	1056.5	1090.6

TABLE 65.—PROPERTIES OF STEAM AT LOW PRESSURES[1].—(*Concluded*)

Temp. °F.	Abs. pressure		Specific volume, cu. ft. per lb.			Total heat or enthalpy, B.t.u. per lb.		
t	Lb. per sq. in. p	In. Hg	Sat. liquid v_f	Evap. v_{fg}	Sat. vapor v_g	Sat. liquid h_f	Evap. h_{fg}	Sat. vapor h_g
67	0.3276	0.6669	0.01605	956.6	956.6	35.05	1056.0	1091.0
68	0.3390	0.6903	0.01605	925.9	925.9	36.04	1055.5	1091.5
69	0.3509	0.7144	0.01605	896.3	896.3	37.04	1054.9	1091.9
70	0.3631	0.7392	0.01606	867.8	867.9	38.04	1054.3	1092.3
71	0.3756	0.7648	0.01606	840.4	840.4	39.04	1053.8	1092.8
72	0.3886	0.7912	0.01606	813.9	813.9	40.04	1053.2	1093.2
73	0.4019	0.8183	0.01606	788.3	788.4	41.03	1052.6	1093.6
74	0.4156	0.8462	0.01606	763.7	763.8	42.03	1052.1	1094.1
75	0.4298	0.8750	0.01607	740.0	740.0	43.03	1051.5	1094.5
76	0.4443	0.9046	0.01607	717.1	717.1	44.03	1050.9	1094.9
77	0.4593	0.9352	0.01607	694.9	694.9	45.02	1050.4	1095.4
78	0.4747	0.9666	0.01607	673.3	673.5	46.02	1049.8	1095.8
79	0.4906	0.9989	0.01608	653.0	653.0	47.02	1049.2	1096.2
80	0.5069	1.0321	0.01608	633.1	633.1	48.02	1048.6	1096.6
81	0.5237	1.0664	0.01608	613.9	613.9	49.02	1048.1	1097.1
82	0.5410	1.1016	0.01608	595.3	595.3	50.01	1047.5	1097.5
83	0.5588	1.1378	0.01609	577.4	577.4	51.01	1046.9	1097.9
84	0.5771	1.1750	0.01609	560.1	560.2	52.01	1046.4	1098.4
85	0.5959	1.2133	0.01609	543.4	543.5	53.00	1045.8	1098.8
86	0.6152	1.2527	0.01609	527.3	527.3	54.00	1045.2	1099.2
87	0.6351	1.2931	0.01610	511.7	511.7	55.00	1044.7	1099.7
88	0.6556	1.3347	0.01610	496.6	496.7	56.00	1044.1	1100.1
89	0.6766	1.3775	0.01610	482.1	482.1	56.99	1043.5	1100.5

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[1] Reprinted, by permission, from "Thermodynamic Properties of Steam," by J. H. Keenan and F. G. Keyes, published by John Wiley & Sons, Inc., 1936.

TABLE 66.—PROPERTIES OF SATURATED WATER VAPOR WITH AIR, 0 TO 104°F.[1]

Temp., °F.	Pressure of saturated vapor		Weight of saturated vapor				Volume, cu. ft. barometer, 29.92 in. hg		Enthalpy per lb.		
	In. Hg	Lb./sq. in.	Per cu. ft.		Per lb. dry air		1 lb. dry air	1 lb. dry air + vapor to saturate it	Dry air, 0°F. datum	Vapor, 32°F. datum	Dry air with vapor to saturate it
			Lb.	Grains	Lb.	Grains					
0	0.03773	0.01853	0.000067914	0.475	0.0007852	5.50	11.58	11.59	0.0000	1059.2	0.8317
1	0.03975	0.01963	0.000071395	0.500	0.0008275	5.79	11.60	11.62	0.2401	1059.7	1.117
2	0.04186	0.02056	0.000075021	0.525	0.0008714	6.10	11.63	11.64	0.4801	1060.1	1.404
3	0.04409	0.02166	0.000078851	0.552	0.0009179	6.43	11.65	11.67	0.7201	1060.6	1.694
4	0.04645	0.02282	0.000082890	0.580	0.0009671	6.77	11.68	11.70	0.9601	1061.0	1.986
5	0.04886	0.02400	0.000087005	0.609	0.001017	7.12	11.70	11.72	1.200	1061.5	2.280
6	0.05144	0.02527	0.000091399	0.640	0.001071	7.50	11.73	11.75	1.440	1061.9	2.577
7	0.05412	0.02658	0.000095955	0.672	0.001127	7.89	11.75	11.77	1.680	1062.4	2.877
8	0.05692	0.02796	0.00010070	0.705	0.001186	8.30	11.78	11.80	1.920	1062.8	3.180
9	0.05988	0.02941	0.00010572	0.740	0.001247	8.73	11.80	11.83	2.160	1063.3	3.486
10	0.06295	0.03092	0.00011090	0.776	0.001311	9.18	11.83	11.85	2.400	1063.7	3.795
11	0.06618	0.03251	0.00011634	0.814	0.001379	9.65	11.86	11.88	2.640	1064.2	4.108
12	0.06958	0.03418	0.00012206	0.854	0.001450	10.15	11.88	11.91	2.880	1064.6	4.424
13	0.07309	0.03590	0.00012794	0.896	0.001523	10.66	11.91	11.93	3.120	1065.1	4.742
14	0.07677	0.03771	0.00013410	0.939	0.001600	11.20	11.93	11.96	3.359	1065.5	5.064
15	0.08067	0.03963	0.00014062	0.984	0.001682	11.77	11.96	11.99	3.599	1066.0	5.392
16	0.08469	0.04160	0.00014732	1.031	0.001766	12.36	11.98	12.01	3.839	1066.4	5.722
17	0.08895	0.04369	0.00015440	1.081	0.001855	12.99	12.00	12.04	4.079	1066.9	6.058
18	0.09337	0.04586	0.00016174	1.132	0.001947	13.63	12.03	12.07	4.319	1067.3	6.397
19	0.09797	0.04812	0.00016935	1.185	0.002043	14.30	12.06	12.09	4.559	1067.8	6.741
20	0.1028	0.05050	0.00017747	1.242	0.002144	15.01	12.08	12.12	4.798	1068.2	7.088
21	0.1078	0.05295	0.00018564	1.299	0.002250	15.75	12.11	12.15	5.038	1068.7	7.443
22	0.1132	0.05560	0.00019439	1.361	0.002361	16.53	12.13	12.18	5.278	1069.1	7.802
23	0.1186	0.05826	0.00020335	1.423	0.002476	17.33	12.16	12.20	5.518	1069.6	8.186
24	0.1244	0.06111	0.00021276	1.489	0.002596	18.17	12.18	12.23	5.758	1070.0	8.536
25	0.1304	0.06405	0.00022255	1.558	0.002722	19.05	12.21	12.26	5.998	1070.5	8.912
26	0.1366	0.06710	0.00023278	1.629	0.002853	19.97	12.23	12.29	6.237	1070.9	9.292
27	0.1432	0.07034	0.00024342	1.704	0.002991	20.94	12.26	12.32	6.477	1071.4	9.682
28	0.1500	0.07368	0.00025445	1.781	0.003133	21.93	12.28	12.34	6.717	1071.8	10.075
29	0.1571	0.07717	0.00026597	1.862	0.003283	22.99	12.31	12.37	6.957	1072.3	10.477
30	0.1645	0.08080	0.00027797	1.946	0.003439	24.07	12.33	12.40	7.197	1072.7	10.886
31	0.1722	0.08458	0.00029043	2.033	0.003601	25.21	12.36	12.43	7.437	1073.2	11.302
32	0.1803	0.08856	0.00030343	2.124	0.003771	26.40	12.38	12.46	7.677	1073.6	11.726
33	0.1879	0.09230	0.00031471	2.203	0.003931	27.52	12.41	12.49	7.917	1074.1	12.139
34	0.1957	0.09610	0.00032690	2.288	0.004094	28.66	12.43	12.51	8.157	1074.5	12.556

TABLE 66.—PROPERTIES OF SATURATED WATER VAPOR WITH AIR, 0 TO 104°F.[1]—(*Concluded*)

Temp., °F.	Pressure of saturated vapor		Weight of saturated vapor				Volume, cu. ft. barometer, 29.92 in. hg		Enthalpy per lb.		
			Per cu. ft.		Per lb. dry air						
	In. Hg	Lb./sq. in.	Lb.	Grains	Lb.	Grains	1 lb. dry air	1 lb. dry air + vapor to saturate it	Dry air, 0°F. datum	Vapor, 32°F. datum	Dry air with vapor to saturate it
35	0.20360	0.1000	0.0003394	2.376	0.004262	29.83	12.46	12.54	8.397	1075.0	12.979
36	0.21195	0.1041	0.0003527	2.469	0.004438	31.07	12.48	12.57	8.636	1075.4	13.409
37	0.22050	0.1083	0.0003662	2.563	0.004618	32.33	12.51	12.60	8.876	1075.9	13.845
38	0.22925	0.1126	0.0003799	2.660	0.004803	33.62	12.53	12.63	9.116	1076.3	14.285
39	0.23842	0.1171	0.0003943	2.760	0.004996	34.97	12.56	12.66	9.356	1076.8	14.736
40	0.24778	0.1217	0.0004090	2.863	0.005194	36.36	12.59	12.69	9.596	1077.2	15.191
41	0.25755	0.1265	0.0004243	2.970	0.005401	37.80	12.61	12.72	9.836	1077.7	15.657
42	0.26773	0.1315	0.0004401	3.081	0.005616	39.31	12.64	12.75	10.08	1078.1	16.13
43	0.27832	0.1367	0.0004566	3.196	0.005840	40.88	12.66	12.78	10.32	1078.6	16.62
44	0.28911	0.1420	0.0004735	3.315	0.006069	42.48	12.69	12.81	10.56	1079.0	17.11
45	0.30031	0.1475	0.0004909	3.436	0.006306	44.14	12.71	12.84	10.80	1079.5	17.61
46	0.31191	0.1532	0.0005088	3.562	0.006553	45.87	12.74	12.87	11.04	1079.9	18.12
47	0.32393	0.1591	0.0005274	3.692	0.006808	47.66	12.76	12.90	11.28	1080.4	18.64
48	0.33635	0.1652	0.0005465	3.826	0.007072	49.50	12.79	12.93	11.52	1080.8	19.16
49	0.34917	0.1715	0.0005663	3.964	0.007345	51.42	12.81	12.96	11.76	1081.3	19.70
50	0.36241	0.1780	0.0005866	4.106	0.007626	53.38	12.84	12.99	12.00	1081.7	20.25
51	0.37625	0.1848	0.0006078	4.255	0.007921	55.45	12.86	13.02	12.23	1082.2	20.80
52	0.39051	0.1918	0.0006296	4.407	0.008226	57.58	12.89	13.06	12.47	1082.6	21.38
53	0.40496	0.1989	0.0006516	4.561	0.008534	59.74	12.91	13.09	12.71	1083.1	21.95
54	0.42003	0.2063	0.0006746	4.722	0.008856	61.99	12.94	13.12	12.95	1083.5	22.55
55	0.43570	0.2140	0.0006984	4.889	0.009192	64.34	12.96	13.15	13.19	1084.0	23.15
56	0.45179	0.2219	0.0007228	5.060	0.009536	66.75	12.99	13.19	13.43	1084.4	23.77
57	0.46828	0.2300	0.0007477	5.234	0.009890	69.23	13.01	13.22	13.67	1084.9	24.40
58	0.48538	0.2384	0.0007735	5.415	0.01026	71.82	13.04	13.25	13.91	1085.3	25.05
59	0.50310	0.2471	0.0008003	5.602	0.01064	74.48	13.06	13.29	14.15	1085.8	25.70
60	0.52142	0.2561	0.0008278	5.795	0.01103	77.21	13.09	13.32	14.39	1086.2	26.37
61	0.54035	0.2654	0.0008562	5.993	0.01144	80.08	13.11	13.35	14.63	1086.7	27.06
62	0.55970	0.2749	0.0008852	6.196	0.01186	83.02	13.14	13.39	14.87	1087.1	27.76
63	0.57985	0.2848	0.0009153	6.407	0.01229	86.03	13.16	13.42	15.11	1087.6	28.48
64	0.60042	0.2949	0.0009460	6.622	0.01274	89.18	13.19	13.46	15.35	1088.0	29.21
65	0.62179	0.3054	0.0009778	6.845	0.01320	92.40	13.21	13.49	15.59	1088.5	29.96
66	0.64367	0.3162	0.0010105	7.074	0.01368	95.76	13.24	13.53	15.83	1088.9	30.73
67	0.66638	0.3273	0.0010440	7.308	0.01417	99.19	13.26	13.57	16.07	1089.4	31.51
68	0.68980	0.3388	0.0010816	7.571	0.01468	102.8	13.29	13.60	16.31	1089.8	32.31
69	0.71382	0.3506	0.0011140	7.798	0.01520	106.4	13.31	13.64	16.55	1090.3	33.12

70	0.73866	0.3628	0.0011507	8.055	0.01574	110.2	13.34	13.68	16.79	1090.7	33.96
71	0.76431	0.3754	0.0011884	8.319	0.01631	114.2	13.37	13.71	17.03	1091.2	34.83
72	0.79058	0.3883	0.0012269	8.588	0.01688	118.2	13.40	13.75	17.27	1091.6	35.70
73	0.81766	0.4016	0.0012667	8.867	0.01748	122.4	13.42	13.79	17.51	1092.1	36.60
74	0.84555	0.4153	0.0013075	9.153	0.01809	126.6	13.44	13.83	17.75	1092.5	37.51
75	0.87448	0.4295	0.0013497	9.448	0.01873	131.1	13.47	13.87	17.99	1093.0	38.46
76	0.90398	0.4440	0.0013927	9.749	0.01938	135.7	13.49	13.91	18.23	1093.4	39.42
77	0.93452	0.4590	0.0014371	10.06	0.02005	140.4	13.52	13.95	18.47	1093.9	40.40
78	0.96588	0.4744	0.0014825	10.38	0.02075	145.3	13.54	13.99	18.71	1094.3	41.42
79	0.99825	0.4903	0.0015295	10.71	0.02147	150.3	13.57	14.03	18.95	1094.8	42.46
80	1.0316	0.5067	0.0015777	11.04	0.02221	155.5	13.59	14.08	19.19	1095.2	43.51
81	1.0661	0.5236	0.0016273	11.39	0.02298	160.9	13.62	14.12	19.43	1095.7	44.61
82	1.1013	0.5409	0.0016781	11.75	0.02377	166.4	13.64	14.16	19.67	1096.1	45.72
83	1.1377	0.5588	0.0017304	12.11	0.02459	172.1	13.67	14.21	19.91	1096.6	46.88
84	1.1752	0.5772	0.0017841	12.49	0.02543	178.0	13.69	14.26	20.15	1097.0	48.05
85	1.2135	0.5960	0.0018389	12.87	0.02629	184.0	13.72	14.30	20.39	1097.5	49.24
86	1.2527	0.6153	0.0018950	13.27	0.02781	190.3	13.74	14.34	20.63	1097.9	50.47
87	1.2933	0.6352	0.0019531	13.67	0.02810	196.7	13.77	14.39	20.87	1098.4	51.74
88	1.3346	0.6555	0.0020116	14.08	0.02904	203.3	13.79	14.44	21.11	1098.8	53.02
89	1.3774	0.6765	0.0020725	14.51	0.03002	210.1	13.82	14.48	21.35	1099.3	54.35
90	1.4211	0.6980	0.0021344	14.94	0.03102	217.1	13.84	14.53	21.59	1099.7	55.70
91	1.4661	0.7201	0.0021982	15.39	0.03205	224.4	13.87	14.58	21.83	1100.2	57.09
92	1.5125	0.7429	0.0022634	15.84	0.03312	231.8	13.89	14.63	22.07	1100.6	58.52
93	1.5600	0.7662	0.0023304	16.31	0.03421	239.5	13.92	14.69	22.32	1101.1	59.99
94	1.6088	0.7902	0.0023992	16.79	0.03535	247.5	13.94	14.73	22.56	1101.5	61.50
95	1.6591	0.8149	0.0024697	17.28	0.03652	255.6	13.97	14.79	22.80	1102.0	63.05
96	1.7108	0.8403	0.0025425	17.80	0.03772	264.0	13.99	14.84	23.04	1102.4	64.62
97	1.7638	0.8663	0.0026164	18.31	0.03896	272.7	14.02	14.90	23.28	1102.9	66.25
98	1.8181	0.8930	0.0026925	18.85	0.04024	281.7	14.04	14.95	23.52	1103.8	67.92
99	1.8741	0.9204	0.0027700	19.19	0.04156	290.9	14.07	15.01	23.76	1103.8	69.63
100	1.9316	0.9487	0.0028506	19.95	0.04293	300.5	14.10	15.07	24.00	1104.2	71.40
101	1.9904	0.9776	0.0029316	20.52	0.04433	310.3	14.12	15.12	24.24	1104.7	73.21
102	2.0507	1.0072	0.0030156	21.11	0.04577	320.4	14.15	15.18	24.48	1105.1	75.06
103	2.1128	1.0377	0.0031017	21.71	0.04726	330.8	14.17	15.25	24.72	1105.6	76.97
104	2.1763	1.0689	0.0031887	22.32	0.04879	341.5	14.20	15.31	24.96	1106.0	78.92

[1] Compiled by W. M. Sawdon, vapor pressures converted from *International Critical Tables.* Copyright, American Society of Heating and Ventilating Engineers—from "Heating-Ventilating-Air Conditioning Guide," 1940, Chap. I.

Note.—This portion of the table covers the range required for comfort air conditioning. For temperatures from 105 to 200°F. see A.S.H. & V.E "Guide."

CHAPTER XIV

PRINCIPLES OF AIR CONDITIONING

231. What Is Air Conditioning?—The term air conditioning has been so widely publicized and is used so loosely that the student should be careful to understand its true meaning. For many years air has been introduced into buildings, after heating, humidifying, and in some cases cooling, and the process has been called ventilation. In recent years much greater stress has been laid on the *treatment* of the air, as compared with the mere operation of circulating it through the building, and this process is now called air conditioning. Air conditioning, then, may be said to be the treatment of the air in buildings so as to render it more comfortable or healthful for human beings or more suitable for manufacturing processes.

The significant factors in air conditioning are many, though unfortunately the term is sometimes used to refer only to certain minor factors such as the cleaning and circulating of air. It is important that the engineer realize exactly what a proposed air-conditioning system does and what it does not do, in order that others may be clearly informed of the results that can be expected from the system.

Complete air conditioning involves the simultaneous control of the following factors:

Temperature
Humidity
Air motion
Air distribution
Dust
Bacteria
Odors
Toxic gases

Of these the first three, temperature, humidity, and air motion, are paramount, and no air-conditioning system deserves the name unless it satisfactorily controls those three factors. Thus it follows that air conditioning embraces heating as well as cooling. Although at the present time considerable emphasis is being laid on artificial cooling and dehumidifying, they are only one phase of the basic problem of temperature and humidity control. In most parts of the United States cooling and dehumidification are required for only a short season, and from the standpoint of hours of use during the year they are of minor importance. They do, however, immeasurably improve comfort in hot weather, and because of that and because their commercial value has become evident, they have rightly come to the fore.

232. Comfort.—The general problem of maintaining comfortable conditions in an occupied room has been the subject of study for many years. Early investigators believed that the atmosphere of a crowded room caused discomfort through its effect on the *lungs*, and carbon dioxide, being the principal product of respiration, was looked upon as the harmful element. Air conditioning, or ventilation, as it was formerly called, was therefore considered for many years a problem of supplying sufficient fresh air to dilute the carbon dixide content.

The celebrated experiments of Leonard Hill were largely responsible for the abandonment of this theory. Hill placed several subjects in an airtight chamber until the carbon dioxide content increased and the oxygen content decreased considerably. They suffered great discomfort; but when the air was set in motion by fans, the discomfort at once disappeared. Other subjects outside the chamber experienced no discomfort from breathing the air from the chamber through tubes, but those within the chamber experienced no relief from breathing outside air. This experiment forcibly demonstrated the modern concept of air conditioning, namely, that the **cooling effect** of the atmosphere upon the skin is of primary importance and that comfort depends upon maintaining the proper thermal environment. The subjects within Hill's test box were uncomfortable because their body heat was not being properly removed, and they became more comfortable as the fans increased the rate of heat removal.

Actually the object of air conditioning for human comfort is not to heat or cool the body, but to control its rate of heat loss within reasonable limits by methods that are known to be most conducive to comfort.

233. Heat Loss from the Human Body.—The human body attempts to maintain itself at a nearly constant internal temperature of about 98.6° while dissipating continually the heat developed from the chemical processes in the utilization of food, drink, and oxygen. To accomplish this, an exceedingly intricate and sensitive control mechanism makes use of several physical and chemical means, such as the variation of the blood supply to the periphery of the body, shivering, and perspiration. Although the temperature-regulating mechanism will compensate for a rather wide range of atmospheric temperature, it is done at the expense of comfort and working efficiency. The conditions of optimum comfort are those under which the temperature-regulating mechanism functions with least effort.

Heat is given off from the body in several ways. The exhaled breath carries off a considerable quantity, and some is dissipated from the surface of the body by radiation, conduction, convection, and the evaporation of perspiration. The total heat emitted from the human body at rest varies but slightly under ordinary indoor conditions, averaging about 400 B.t.u. per hr. for adult men. The amount of heat lost by radiation and convection decreases rapidly as the dry-bulb temperature rises, whereas that lost by evaporation is correspondingly increased as the temperature-regulating mechanism causes more perspiration to be exuded. Heat loss from the human body by conduction is negligible unless the body is in contact with a colder surface.

Practically the entire surface of the body is normally in contact with air, either with the atmosphere directly (hands and head) or with an intermediate layer of air inside the clothing. Thus the loss of heat from the body takes place principally by evaporation and convection.

Under normal conditions, in heated rooms, radiation usually accounts for 46 to 60 per cent of the total heat loss; convection, for 15 to 30 per cent; and evaporation, for 20 to 30 per cent. The proportions depend upon the amount of clothing and upon the wall temperatures and other physical conditions of the environment.[1]

234. Factors Determining Comfort.—There are four factors which, by their combined effect, determine the cooling power of the atmosphere upon the human body. They are **temperature,**

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[1] Winslow, Herrington, and Gagge, "The Determination of Radiation and Convection Exchanges by Partitional Calorimetry," *Am. Jour. Physiol.*, August, 1936.

humidity, air motion, and radiation. The dry-bulb temperature of the atmosphere has long been used alone as an index of comfort, but it is obviously an incomplete index. The humidity and the amount of air motion are of importance. Radiation to cold or from warm surfaces is another important factor under some circumstances.

The effects of humidity and of air movement are matters of common experience. When the indoor air is very dry, a higher temperature is necessary for comfort because of the greater cooling power of the dry air. The evaporation of the perspiration with which the body is always partially covered has a marked influence upon the rate of heat loss, and the rate of evaporation is dependent upon the humidity of the surrounding atmosphere.

The cooling action of air motion upon the skin is well known. It increases heat removal by the increased evaporative effect and by an increase in the convection effect as the layer of air in contact with the skin is repeatedly renewed.

Radiation to and from the body is an important factor in comfort, but probably because it is difficult to measure and control it has not been given the attention in heating and air-conditioning work that it deserves.

Loss of heat by radiation from the exposed body and clothing depends on the temperature of the surrounding walls and is independent of air temperature. Air movement reduces to some extent the radiation loss by lowering the surface temperature of the body. The rate of heat loss varies according to Stefan's fourth-power law [Eq. (7), Chap. I] and is dependent upon the difference in temperature between the surface of the body and the mean surface temperature of the environment. This latter temperature is referred to as the *mean radiant temperature,* and a method of evaluating it is outlined in Art. 78.

It is possible to maintain comfortable conditions at relatively low air temperatures by utilizing the principle of panel heating discussed in Chap. V, whereby the mean radiant temperature of the room surface is maintained a few degrees above the air temperature. Likewise it is possible, though of doubtful practicability, to maintain comfort at high air temperatures by means of cold surrounding surfaces. In the first case it is likely that the *normal* radiation loss of 50 per cent from the human body is closely approached. In the latter case, however, it is probable that the resulting high radiation loss causes an *abnormal* adjustment in the heat-regulating mechanism of the body in order to maintain comfort.

For example, if the surface temperatures of the surrounding walls and glass are substantially lower than the air temperature, the amount of heat loss by radiation is increased, while the amount lost by convection and evaporation is reduced. To compensate for this condition some rooms must be maintained around 75° in *winter* to offset the increased radiation loss from the human body to the cold walls, unless adequate wall insulation and double windows are provided. Conversely, in *summer* the room air temperature must be lower for a given sensation of comfort if the walls, ceiling, and floor are at a high temperature than if they are at a low temperature.

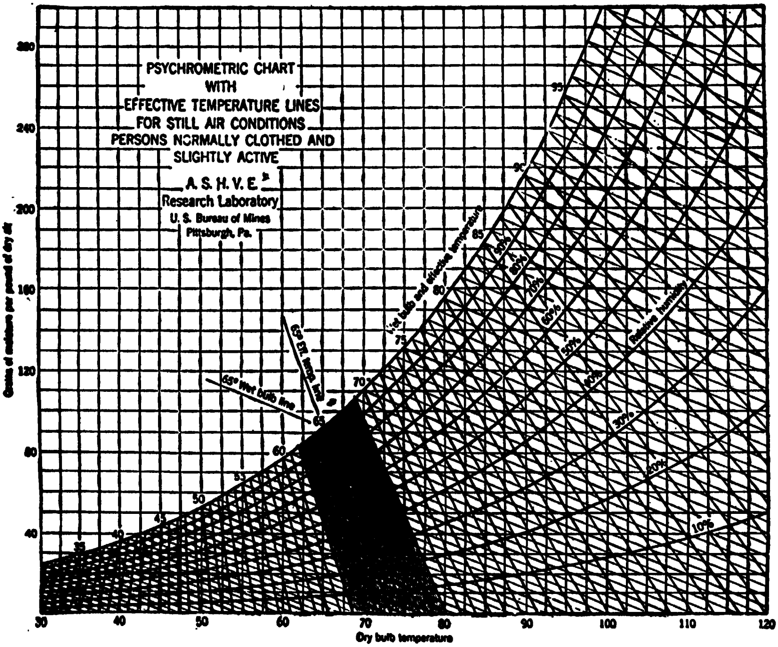

Fig. 196.[1]

235. Effective Temperature.—Experiments made on a large number of human subjects have demonstrated that there is a consistent relation between the temperature, humidity, and air motion in their effect upon comfort.[1] When the results of such observations are plotted on a psychrometric chart it is found

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[1] Copyright, 1929, A.S.H. & V.E.

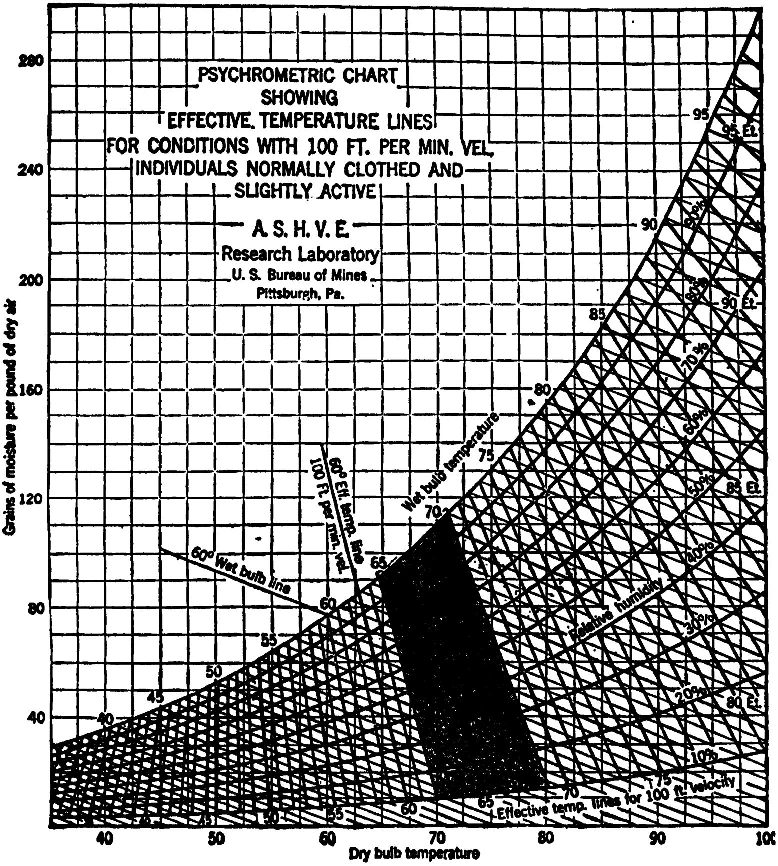

FIG. 197.[2]

that the points representing conditions of equal warmth fall in approximately straight lines. In Fig. 196 is shown a psychrometric chart on which these lines are superimposed. They are called lines of **effective temperature,** and the effective temperature *in still air* of any point on the psychrometric chart is the wet-bulb temperature corresponding to the intersection of the effective-temperature line through that point with the saturation curve.

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[1] The detailed results of these extended investigations made at the Research Laboratory of the A.S.H. & V.E., in cooperation with the U. S. Bureau of Mines and the U S. Public Health Service, are reported in various papers in the *Transactions* of that society, from 1920 to 1928.

[2] Copyright, 1929, A.S.H. & V.E.

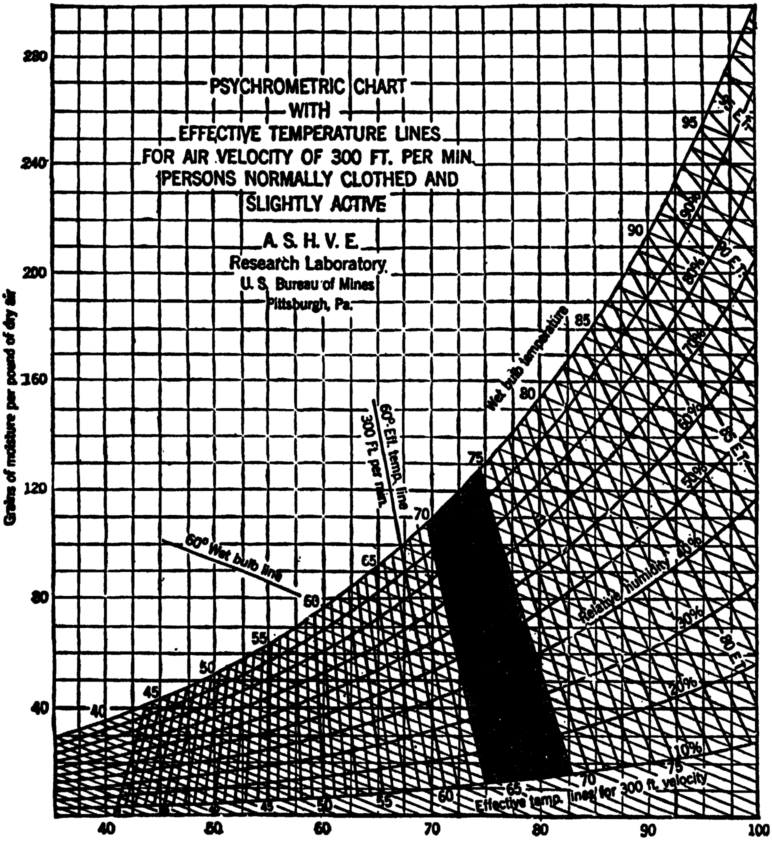

Fig. 198.[1]

Thus in Fig. 196 the effective temperature for a condition of 59° wet bulb and 68° dry bulb is 65°, and the same effective temperature corresponds to 48° wet bulb and 72.5° dry bulb. This means that a person would feel equally warm under either of these conditions.

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[1] Copyright, 1929, A.S.H. & V.E.

The effective-temperature scale for still air was arbitrarily made to correspond with the scale of wet-bulb (and dry-bulb) temperatures on the saturation line. For *moving air* the effective temperature for a given temperature and humidity is lower, and the two scales cannot coincide. In Figs. 197, 198, and 199 the effective temperatures are shown for air velocities of 100, 300, and 500 ft. per min., the effective-temperature scale being shown near the bottom of the chart.

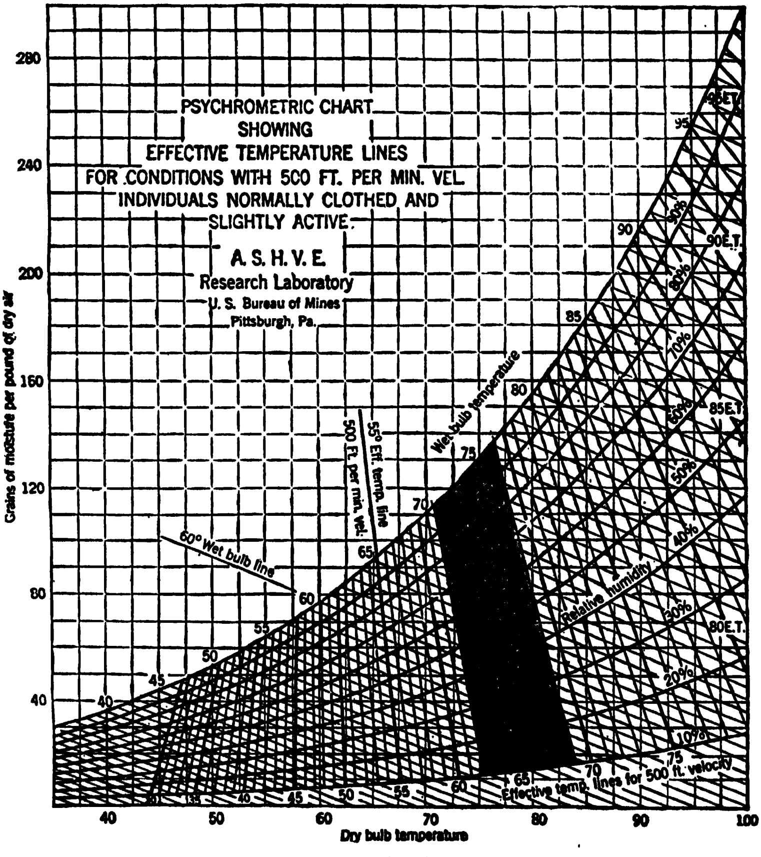

Fig. 199.[1]

The effective temperature is the only true index of human comfort that is now available. Neither the wet-bulb nor the dry-bulb thermometer alone truly indicates the condition of the air from a comfort standpoint.

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[1] Copyright, 1929, A.S.H. & V.E.

Note that in the psychrometric charts of Figs. 196 to 199 the effective temperature lines become vertical at certain dry-bulb temperatures and that their direction of slope then reverses. In Fig. 196 the change of slope occurs at about 45°. At higher temperatures (for still air) an increase in humidity causes an increase in the effective temperature, whereas with lower temperatures an increase in humidity causes a decrease in the effective temperature. This corresponds with our experience with the greater chilling effect of damp air at low temperatures. With moving air the change takes place at a higher temperature and is more marked. At 100, 300, and 500 ft. per min. the critical temperatures are respectively, 51, 56, and 59°.

The relative importance of changes in temperature and humidity should be observed as they affect the effective temperature. For example, an effective temperature of 65° in still air can be attained by a temperature of 67° with a relative humidity of 70 per cent or by a temperature of 70.5° with a relative humidity of 30 per cent. Thus an increase of temperature of 3.5° compensates for a decrease in humidity of 40 per cent. In other words, 1° difference in temperature is more significant than a 10 per cent difference in relative humidity. Changes in humidity do affect comfort, of course, but their importance is minor compared to temperature changes, and small variations are insignificant. This is important from a standpoint of temperature control for both summer and winter conditions. It makes it possible to maintain satisfactorily comfortable conditions by controlling the dry-bulb temperature alone, allowing the humidity to vary as it may over a reasonable range.

The effect of air motion is more pronounced. For an effective temperature of 65° the following dry-bulb temperatures correspond to the various air velocities:

Still air	69° dry bulb
100 ft. per min	71.7° dry bulb
300 ft. per min	73.5° dry bulb
500 ft. per min	77.0° dry bulb

In other words, to compensate for an air movement of 300 ft. per min. an increase in dry-bulb temperature of 4.5°, as compared with still air, is necessary. Actually the effect of air movement in the form of a local draft is difficult to neutralize by an increase in temperature, and for people at rest an air movement over 50 ft. per min., or sometimes even less, will cause discomfort.

236. Comfort Zones.—The shaded zones in the effective-temperature charts are the zones of comfort, in which 50 per cent or more of the people who were the subjects in the tests expressed themselves as feeling comfortable. It will be noted that the zones extend from the saturation line to a humidity of 10 per cent or, in the still-air chart, to zero humidity. It is recognized that extremely high or low humidities are not comfortable, and these zones would be more properly termed zones of *acceptable warmth.* More recent consideration of this point indicates that the true *comfort* zones should be much more restricted, and the range between 30 and 70 per cent humidity[1] has been recommended. The *optimum* humidity conditions probably cover a much narrower range, and practical considerations such as the possibility of condensation on walls and windows would limit the humidity, in winter, to much below 70 per cent.

Another factor affecting the permissible relative humidity is the point at which sensible perspiration begins. In the work by Houghten and others, previously referred to, it was found that at 95 per cent relative humidity an effective temperature of 73° was sufficient to produce on the foreheads of 40 per cent of the subjects a feeling of clamminess that marks the beginning of perspiration. At 20 per cent humidity the corresponding effective temperature was 75°. The point to be remembered is that the higher levels of relative humidity may cause discomfort from perspiration, even though the conditions of *warmth* may be satisfactory.

The foregoing refers to *sensible* perspiration. There is a certain amount of insensible perspiration at temperatures below those at which actual sweating begins,[2] and that moisture does not emanate from the sweat glands but from the capillaries.

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[1] See report of committee headed by C. P. Yaglou, "How to Use the Effective Temperature Index and Comfort Charts," *Trans. A.S.H. & V.E.*, 1932, and also, "Comfort with Summer Air Conditioning," *Trans A.S.H. & V.E.*, 1942.

[2] HOUGHTEN, TEAGUE, MILLER, and GANT, "Thermal Exchanges between the Human Body and Its Environment," *Am. Jour. Physiol.*, April, 1929.

237. Limitations of the Effective-temperature Charts.—It is important to bear in mind the conditions under which the effective-temperature charts in Figs. 196 to 199 were obtained, because they apply only to similar conditions. The experiments were made in rooms having no sources of radiant heat, and the walls of the rooms were substantially at the air temperature. The winter condition in most rooms only roughly approximates this, as there is often some radiation from heating units and usually some cold wall or window surfaces to which the bodies of the occupants radiate heat. The charts would not apply to conditions in which much or all of the heat was supplied by radiation, as it is in some buildings.

The subjects in the experiments on which this set of charts are based wore normal winter clothing. A different set of charts, known as the basic charts, were constructed from tests on subjects stripped to the waist. They are useful in certain industries involving very hot conditions.

The charts are applicable to persons in the northeastern part of the United States between the ages of twenty and seventy years, normally clothed and slightly active.

In an effort to evaluate the radiation effect some preliminary tests[1] were conducted within an enclosure and the results indicate that for each 1° elevation or depression of the *mean radiant temperature* above or below the air temperature, about 0.5° counterchange in *effective temperature* is required. Additional observations will be required to verify this tentative finding, which is one of the livest subjects in air-conditioning research at present.

238. The Summer Comfort Zone.—Yaglou and Drinker[2] have investigated the relation between the comfort zones for summer and winter, and Fig. 200 shows the results of their extended observations on subjects placed in a test room in summer, as compared with similar results obtained in winter at the A.S.H. & V.E. laboratory. It will be observed that the line of effective temperature at which the maximum number of people were most comfortable is 66° for the winter comfort zone and 71° for the summer comfort zone, the air movement being 15 to 25 ft. per min. The marked difference may be accounted for partly by acclimatization, people becoming accustomed to higher temperatures in summer, and partly by differences in clothing.

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[1] HOUGHTEN, GUNST, and SUCIU, "Radiation as a Factor in the Sensation of Warmth," *Trans. A.S.H. & V.E.*, 1941.

[2] YAGLOU and DRINKER, "The Summer Comfort Zone: Climate and Clothing; A Report of Work Done at the Harvard School of Public Health," *Trans. A.S.H. & V.E.*, 1929.

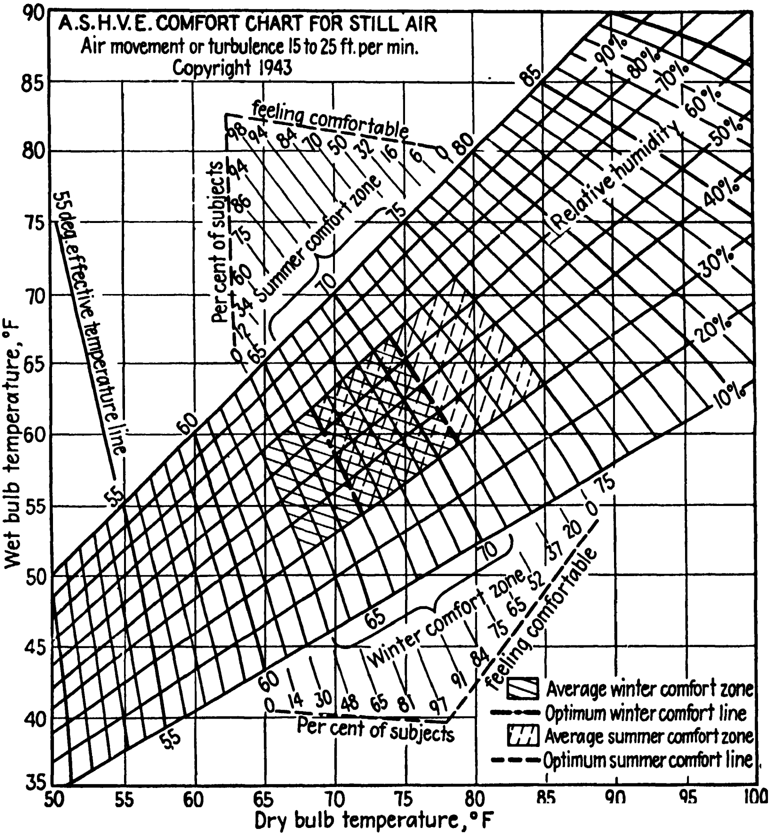

FIG. 200.—Effective temperature chart showing summer and winter comfort zones.[1]

The question as to the most desirable conditions to be maintained in summer in air-conditioned buildings is not yet satisfactorily answered. In the early years of summer comfort cooling, many buildings were grossly overcooled, and the entire air conditioning was widely discredited. There are still instances of overcooling, but operators generally have come to realize that although 70° may be comfortable with winter conditions and winter clothing, it is not comfortable in a theater or store in summer.

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[1] From *Trans.*, Vol. 38, 1932; copyright, *A.S.H. & V.E.*

The present question as to optimum conditions is not that of making people comfortable while in the air-conditioned space for a long period. That point is fairly well covered, at least for the Northern states, by the comfort zone of Fig. 200. It concerns rather the matter of maintaining such conditions indoors that people entering and leaving the building will not be subjected to serious discomfort. The temperature-regulating mechanism of the body cannot function instantaneously. A person entering a cool, dry atmosphere while bathed in perspiration, which his body needed outdoors for its evaporative cooling effect, suffers an appreciable shock by reason of the sudden and enormous increase in the rates of evaporation, convection, and conduction. Conversely, as he leaves the building the rate of heat loss from the body is temporarily reduced in a drastic manner.

The problem, then, is to select such indoor conditions as will be reasonably comfortable but still not too far below those of the out-of-doors. In a type of building such as a bank or a store, assuming that it is the comfort of the customers that is sought primarily, the difference between the indoor and outdoor temperatures can be made rather small and will then be satisfactory. But in an office building, for example, where most of the occupants are inside for several hours but others are entering and leaving, it is practically impossible to keep the long-time occupants comfortable without causing some degree of shock to them as they leave and to the transients. The same is true in a theater.

Table 67 shows the indoor effective temperatures corresponding to various outdoor temperatures as specified for *design conditions* in the A.S.H. & V.E. Code of Minimum Requirements for Comfort Air Conditioning. From the standpoint of reducing the entering and leaving shock, the conditions specified are probably a reasonable compromise; but for long-time occupancy effective temperature of over 73° will be uncomfortably warm for most people regardless of the outside conditions.

Control devices have been developed to vary the indoor temperature automatically in relation to the outdoor temperature,

TABLE 67.—DESIGN CONDITIONS FOR AIR-CONDITIONING SYSTEMS[1]

Design Outside Dry-bulb Temperature, °F	Design Inside Effective Temperature, °F.
105	75.5
100	75
95	74
90	73
85	72
80	71

[1] A.S.H. & V.E. Code of Minimum Requirements for Comfort Air Conditioning, *Trans. A.S.H. & V.E.*, 1938.

and many systems are so operated. While the outdoor temperature is rising during the day, the differential relationship can be maintained without difficulty; but if the outdoor temperature falls suddenly, as because of a sudden storm, the system is unlikely to have sufficient capacity to reduce the inside temperature so rapidly.

There are many who advocate a moderate indoor temperature, of 72° effective temperature or thereabouts, to be maintained continuously for buildings such as office buildings in cities where outdoor dry-bulb temperature seldom exceeds 95°.

For purely short-time occupancy, as in a store, the indoor temperatures of Table 67 are fairly satisfactory but perhaps somewhat too low.[1]

Much study is being given to this whole question, and valuable information is being contributed.[2]

239. Metabolism.—The actual amounts of heat and moisture given off from the human body are important because they affect the design of the air-conditioning system. In winter the heat and moisture from the occupants may in some cases reduce the amounts that must be supplied artificially; but in summer they are often an important item of load on the cooling and dehumidifying systems. For the latter application it is necessary to separate the total heat loss into the amount of heat given off as sensible heat and the amount given off as latent heat in the evaporation of moisture.

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[1] See J. H. WALKER, "How Cool? Inside Temperature Should Depend upon Type of Occupancy," *Heating and Ventilating*, October, 1932.

[2] HOUGHTEN and GUTBERLET, "Comfort Standards for Summer Air Conditioning," *Trans. A.S.H. & V.E.*, 1936. HOUGHTEN, GIESECKE, TASKER, and GUTBERLET, "Cooling Requirements for Summer Air Conditioning," *Trans. A.S.H. & V.E.*, 1937. NEWTON, HOUGHTEN, GUTBERLET, QUALLEY, and TOMLINSON, "Shock Experiences of 275 Workers after Entering and Leaving Cooled and Air-conditioned Offices," *Trans. A.S.H. & V.E.*, July, 1938.

One of the fundamental processes of the human body is that of metabolism, by which the elements in the food—principally carbon and hydrogen—unite with the oxygen taken into the lungs, to develop body heat and to produce energy for internal and external work. When the body is at rest the metabolic rate is at a minimum, but when external work is performed the metabolic rate increases markedly.

As a heat engine the human body is inefficient. Tests have shown that in some cases as little as 11 per cent of the energy corresponding to the total metabolic rate was actually converted into work. In most cases the energy applied to a given operation is converted into heat within the room which is to be air conditioned, and this fact coupled with the low efficiency of the process of metabolism makes it permissible to take the total metabolism, expressed in heat units, as the amount of heat dissipated. If the operation is such that the rate of doing work is known, the corresponding metabolic rate may be calculated by assuming an efficiency of 20 per cent and adding to this metabolic rate equivalent to the work the metabolic rate of the body at rest under the given conditions to give the *total* metabolic rate.

240. Heat and Moisture Loss.—Extended studies have been made of heat and moisture loss from the human body under various atmospheric conditions and various degrees of activity.[1] It was found that the total heat loss for men at rest is approximately 400 B.t.u. per hr. for effective temperatures between 65° and 87°. It increases below 65° and decreases rather sharply above 85°. This relation is shown in Fig. 201. It illustrates how the body accelerates or retards its production of heat in the effort to maintain a uniform body temperature.

The two components of this total heat loss are the sensible heat loss, which takes place by radiation and convection, and the

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[1] See HOUGHTEN, TEAGUE, MILLER, and YANT, "Heat and Moisture Losses from the Human Body and Their Relation to Air Conditioning Problems," *Trans.* A.S.H. & V.E., 1929; "Heat and Moisture Losses from Men at Work and Application to Air Conditioning Problems," *Trans. A.S.H. & V.E.*, 1931. Figures 201, 202, and 203 are taken from these papers.

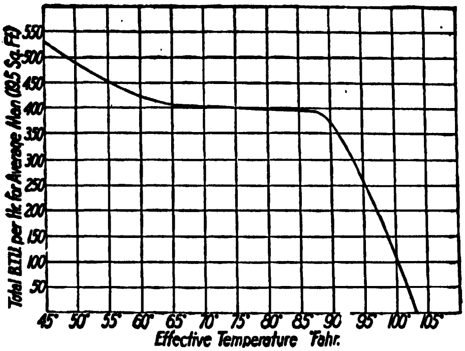

FIG. 201.—Heat given off per hour by the average man in relation to effective temperature.

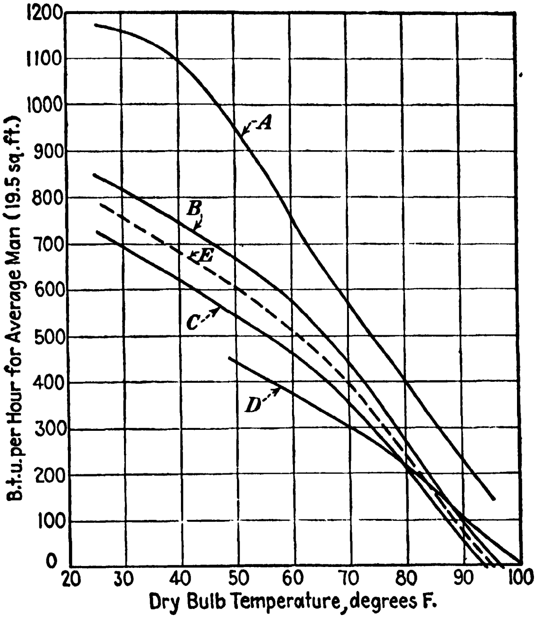

FIG. 202.—Relation between sensible heat loss from the human body and dry-bulb temperature for still air.[1]

Curve *A*—Men working 66,150 ft.-lb. per hr. Curve *B*—Men working 33,075 ft.-lb. per hr. Curve *C*—Men working 16,538 ft.-lb. per hr. Curve *D*—Men seated at rest. Curves *A* and *C* drawn from data at a dry-bulb temperature of 81.3°F only and extrapolating the relation between curves *B* and *D*, which were drawn from data at many temperatures.

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[1] See footnote, p. 350.

latent heat loss, which goes to evaporate perspiration and respiratory moisture. These two items of heat loss vary differently as the environment is changed. The sensible heat loss is governed by the physical laws of heat transfer from a surface to a surrounding gas and is largely a function of the dry-bulb temperature. It decreases with increasing temperature and becomes zero when the air temperature exceeds the body temperature.

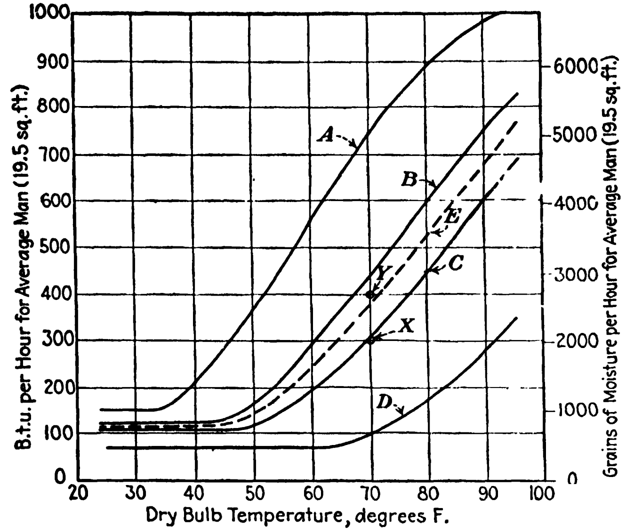

FIG. 203.—Latent heat and moisture loss from the human body by evaporation, in relation to dry-bulb temperature for still-air conditions.[1]

Curve *A*—Men working 66,150 ft.-lb. per hr. Curve *B*—Men working 33,075 ft.-lb. per hr. Curve *C*—Men working 16,538 ft.-lb. per hr. Curve *D*—Men seated at rest. Curves *A* and *C* drawn from data at a dry-bulb temperature of 81.3°F. only and extrapolating the relation between curves *B* and *D*, which were drawn from data at many temperatures.

Heat loss by evaporation, or latent heat, is controlled by the temperature-regulating mechanism of the body through its control of the rate of perspiration. It is affected by air motion and by the humidity of the atmosphere. The weight of moisture evaporated bears a fixed relation to this portion of the heat

The curves of Fig. 202 give the sensible heat loss for men at rest and doing work at various rates, and those of Fig. 203

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[1] See footnote p. 350.

give the latent heat and moisture loss. From these curves essential information for the design of cooling and dehumidifying systems can be obtained.

Example.—Determine the sensible heat, latent heat, and moisture loss from 1,200 people seated in an auditorium in which the dry-bulb temperature is 72°.

From Fig. 202, curve *D*, the sensible heat loss from men at rest at 72° is 287 B.t.u. per hr. For 1,200 persons it would be 344,400 B.t.u. per hr.

From Fig. 203, curve *D*, the latent heat loss is 110 B.t.u., and the moisture loss 780 gr. at 72°. For 1,200 people the latent heat loss would be 132,000 B.t.u. per hr., and the moisture loss 134 lb. per hr.

It is assumed that all the latent heat for the evaporation of moisture is produced by the body. This is not entirely correct, because, at high temperatures at least, some of the latent heat comes from the air and therefore does not become a load upon the cooling system. The studies that have been made have not attempted to solve this rather fine-drawn question, however, and for engineering purposes it is sufficiently accurate to assume that all the latent heat comes from the bodies of the occupants.

The metabolic rate increases sharply with a slight degree of activity. The following figures illustrate this point:[1]

	Heat dissipated, B.t.u. per hour		
	Sensible heat	Latent heat	Total
Seated, at rest	225	159	384
Standing, at rest	225	206	431
Office worker, moderately active	225	265	490

241. Humidity and Air Motion.—Although the comfort zones indicate a wide range of conditions under which people are equally comfortable, there are other factors that must be considered in determining the *best* conditions.

A moderate humidity is desirable, neither so high as to be unpleasant nor so low as to cause excessive dryness of the mucous membranes of the respiratory tract. There is much disagreement as to the latter. Although, as rightly pointed out by various investigators,[1] there is little actual evidence that low humidities are detrimental to health, there are many physicians who hold that excessive dryness is harmful and increases the liability of infection. There is also evidence that many people believe that they are more comfortable in a moderately moist atmosphere, although this view is not universal.

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[1] For additional data see A.S.H. & V.E., "Guide," 1945.

Therefore, from a standpoint of comfort and perhaps of health, there is a rather widely accepted view, admittedly not based on scientific evidence, that a moderate humidity (with a minimum of 25 or 30 per cent) should be maintained.

There are other advantages of moderate humidity, apart from comfort or health considerations. Everyone knows how woodwork shrinks in winter and swells in summer with uncontrolled humidity. But the maintaining of artificial humidities in houses not properly designed for them may, as discussed in Chap. II, be a source of trouble.

A certain amount of air motion is also to be desired. In still air the body is enveloped by a layer of warm and humid air, which a moderate air movement helps to blow away. An air movement that is just perceptible has a stimulating action, probably through its effect upon the sensory nerves; but too great a velocity may cause extreme discomfort, and the following precautions regarding air motion must be rigidly observed in the design of an air-conditioning system if satisfactory results are to be obtained:

1. The maximum air velocity that can be tolerated with comfort by human beings at rest is approximately 50 ft. per min. Even less is preferable.
2. Air currents of appreciable velocity should not be directed downward upon the occupants of a room nor allowed to strike them from behind.
3. There should be no high-velocity jets of air directed into the occupied zone.
4. Air should not be introduced at a temperature too far below that of the room, especially in winter.

Improper or excessive air movement is probably the most frequent cause of complaint regarding ventilating systems. It is not always due to the design of the system but often to its adjustment or to careless operation.

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[1] For example, C. P. YAGLOU, "Physical and Physiological Principles of Air Conditioning," *Jour. Am. Med. Assoc.*, May and September, 1937.

Another desirable feature, but one concerning which little is known at present, is that of *variability*, or freedom from monotony. Variations in the degree and direction of air movement (and in the air temperature) are known to have a refreshing effect; but the permissible limits and the practical value of this are yet to be studied.

The measurement of air motion is somewhat difficult at the low velocities existing in rooms. The ordinary anemometer is quite useless. One method is to produce puffs of smoke or visible vapor and time their movement. Tobacco smoke or the white vapor resulting from the mixing of ammonia and hydrochloric acid fumes may be used. The katathermometer devised by Leonard Hill to measure the cooling power of the atmosphere may also be used for the measurement of air motion.[1] Other devices are the hot-wire anemometer and certain new forms of vane anemometer (See Chap. XVII).

The permissible air velocity is related to the dry-bulb temperature. Studies[2] indicate that for drafts striking the back of the neck a velocity of 60 ft. per min. for a temperature of 70° and 40 ft. per min. for a temperature of 69° represents the border line of discomfort. For drafts on the ankles 55 ft. per min. for 67° temperature and 37 ft. per min. for 66° were the limiting conditions.

242. Atmospheric Pollution.—The factors to be considered in ventilation, next in importance to the ones that have been discussed, are those relating to the **purity** of the atmosphere. The air in buildings receives many polluting elements, and satisfactory air conditioning requires that they be kept within limits. The most common sources of pollution are the products of respiration, namely, carbon dioxide, moisture, and odors. Carbon dixoide, as has been stated, was formerly regarded as definitely harmful but is now thought to be quite harmless.

The average person breathes at the rate of about 17 respirations per minute while at rest. At each respiration about 30½ cu. in. of air is inhaled, or about 18 cu. ft. per hr., which amounts to about 34 lb. of air in 24 hr., or a little over 7 lb. of oxygen. The inhaled air loses about 5 per cent of its oxygen content while in the lungs and gains from 3.5 to 4 per cent of carbon dioxide. The percentage composition of free air and of expired air, by volume, is about as follows:

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[1] HILL, VERNON, and ASH, "The Kata-thermometer as a Measure of Ventilation," *Proc. Roy. Soc. (London)*, Vol. 93.

[2] HOUGHTEN, GUTBERLET, and WITKOWSKI, "Draft Temperatures and Velocities in Relation to Skin Temperatures and Feeling of Warmth," *Trans. A.S.H. & V.E.*, 1938.

Components of air	Free atmosphere, per cent (approximately)	Expired air, per cent (approximately)
Oxygen	20.9	15.4
Nitrogen	79.1	79.2
Carbon dioxide	0.03 to 0.04	4.03 to 4.04

243. Odors.—Formerly the significance of odors in ventilation was not appreciated, but odors are now recognized as being extremely important, especially as to the first impressions produced as a person enters a room. Many complaints of poor ventilation can be directly traced to the presence of unpleasant odors, sometimes of very obscure origin. In fact, it may be said that, next to the cooling power of the atmosphere as indicated by the effective temperature, odors are the most important consideration from the standpoint of producing results satisfactory to the occupants.

Odors are sometimes objectionable in that they indicate the presence of genuinely harmful substances, but the chief effect of odors is usually a psychological one. The most common odors in an occupied room emanate from the human body—from the mouth, throat, and lungs and from perspiration and soiled clothing. This so-called "crowd smell" is not truly harmful, for it has been demonstrated to have little effect upon health; but modern standards of cleanliness and comfort require a reasonably wholesome atmosphere, and the reduction of odors in crowded rooms is one of the most important problems in air conditioning.

The odors from furnishings, rugs, and hangings, though nearly imperceptible, may impart a feeling of stuffiness to the atmosphere. Such odors are often very difficult to trace to their source.

When conditions exist where it is necessary to remove odors, it is possible to absorb them in **activated carbon.** Air washers should not be expected to deodorize air effectively, as few of the particles causing odor are soluble in water. Some odor problems may be satisfactorily solved by the dilution method, in which increased percentages of fresh air are introduced to the space being ventilated.

No accurate standard has been found for the measurement of odors. One method is to compare the odor in the room with a number of odoriferous solutions of varying intensities.

244. Dust.—The air, especially in cities, contains a large amount of dust of various degrees of fineness and consisting of many different substances. The many tons of cinders, soot, and ash particles cast out into the air from chimneys constitute no small part of the total. Other constituents are the products of abrasion of pavements and rubber tires, organic matter of various kinds, and a great many other materials.

Industrial operations often produce particular kinds of dust, some of which, because of their sharpness, chemical activity, or infectious nature, may be quite injurious when breathed.

Drinker[1] suggests a classification of atmospheric impurities based on the size of the particles as follows:

1. **Dusts.**—Particles of 150 to 1 micron[2] in diameter. Such particles are thrown into the air by mechanical agencies, as in grinding, crushing, drilling, and blasting.
2. **Fumes.**—Particles of 0.2 to 1 micron, resulting from reactions such as distillation, complete and incomplete oxidation of metal fumes, and purely chemical reactions. *Examples:* ammonium chloride, lead and mercury fumes, zinc oxide, sulphuric acid mist.
3. **Smokes.**—Particles less than 0.3 micron, usually resulting from incomplete combustion of carbonaceous material such as coal, oil, tar, and tobacco.

The enormous range in size and the extreme fineness of the smaller particles are evident from this classification. In practice the larger particles, such as the true dusts from abrasive operations, are very much the easiest to filter or wash from the air. Fine smokes, such as tobacco smoke, will penetrate even the finest of filters to some extent. Although actually consisting of solid particles, they behave very much like a gas and cannot be removed by a gravity-settling process.

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[1] *Trans.* A.S.H. & V.E., p. 52, 1925.

[2] A micron is one-millionth of a meter, or 1/25,000 of an inch.

The **measurement** of air dustiness has engaged the attention of many research workers but cannot be said to have been solved with entire satisfaction, nor are there generally recognized standards as to what the permissible dustiness of air should be.

There are several methods of collecting dust from a measured volume of air, chief among them being the porous-cup method, the sticky-plate-counting method, and the electric-precipitator method. The porous-cup method is specified in the filter test code of the A.S.H. & V.E. (see Chap. XVI).

Pollen from flowers and weeds is a serious impurity in the air, from the standpoint of the hay-fever sufferer. Adequate filters in an air-conditioning system are reasonably effective in reducing the pollen count, and in many hay-fever cases the patient is very much more comfortable in a properly air-conditioned room.

245. Bacteria.—The determination of bacteria count is a task for the bacteriologist or one skilled in such work. The instrument that is regarded as most suitable for determining the bacteria in the atmosphere is the Wells air centrifuge. This instrument draws in air at a measured rate of flow and precipitates the bacteria onto the inner surface of a glass bottle coated with agar. After incubation they are counted.

No rigid limits have been established for permissible bacteria content, but it is recognized that the degree of air contamination may be reduced by proper ventilation. Table 68 shows average values of bacteria count from a large number of observations in several locations in New York City.

246. Air Sterilization.—Continued efforts to diminish the amount of fresh air supplied for ventilation and thereby to reduce the heating or cooling load has led to a growing interest in sanitary ventilation. Wells[1] has shown that aerial infection by pathogenic bacteria beyond the limits of the mouth spray takes place in occupied rooms. These air-borne bacteria may remain suspended in the air long enough to be transmitted to other persons in the same room or building.

The possibility of sterilizing air with **ultraviolet light**[2] by destroying the bacteria at their point of admission to the air shows some promise. Even direct sunlight that has passed through window glass is bactericidal to a varying degree. However, the ultraviolet range shown in Fig. 1 of Chap. I is the most potent for air purification.

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[1] W. F. WELLS and M. W. WELLS, "Air-borne Infection," *Jour. Am. Med. Assoc.*, 1936.

[2] W. F. Wells, "Air Disinfection in Ventilation," *Jour. A.S.H. & V.E.*, June, 1944.

The idea of employing bactericidal mists[1] for controlling air-borne infection is now being studied. These mists consist of tiny droplets of either **propylene glycol** or **triethylene glycol,** and are dispersed in the air in concentrations of 1 gram of glycol to from 50 to 200 million cu. cm. of air. Complete data regarding the practical application of germicidal mists to air conditioning are still lacking.

TABLE 68.—BACTERIA PER CUBIC FOOT OF AIR AT VARIOUS LOCATIONS[1]
(Incubated 18 hr. at 37°C., week ending Friday, Jan. 10, to week ending Friday, June 26, 1936)

Type of location	Sheep's-blood beef-infusion agar		Beef-infusion agar	
	No. of samples	Bacteria per cu. ft. of air	No. of samples	Bacteria per cu ft. of air
Schools	707	29.6	678	21.1
Subway	290	19.2	294	12.4
Theater, nonventilated (auditorium)	104	13.2	110	8.4
ventilated (ducts)	149	3.1	114	1.4
Streets	143	11.2	134	8.5
Park	13	3.0	18	1.8

[1] PINCUS and STERN, *Can. Pub. Health Jour.*, April, 1937.

247. Air Supply.—Although it will be seen from the foregoing that the quality rather than the quantity of the air supply is the important consideration, nevertheless a sufficient supply of properly distributed air is essential. In rooms occupied by many people the desired air conditions cannot be obtained without a reasonably generous air supply. On the other hand, the permissible quantity is limited by the danger of creating an uncomfortably high rate of air movement.

In many states there are laws governing the minimum amount of air that must be supplied for various kinds of buildings.

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[1] ROBERTSON and others, "Bactericidal Action of Propylene Glycol Vapor on Micro-organisms Suspended in Air," *Jour. Exp. Med.*, 1942; also "The Lethal Effect of Triethylene Glycol Vapor on Air-borne Bacteria and Influenza Virus," *Science*, 97: 142, 1943.

The figure usually specified for schools and auditoriums is 30 cu. ft. per min. per person.

The quantities of air that represent good practice for various kinds of buildings are given in Table 69. These quantities in most cases mean the amount of new air drawn from outside; but experience seems to show that from 50 to as much as 75 per cent may safely be recirculated, especially if it is passed through an air washer or other air-purifying apparatus. The present tendency is to permit a considerable amount of recirculation but to require a minimum of 10 cu. ft. per min. of fresh air per occupant. Air from kitchens, toilet rooms, etc., should never be recirculated.

TABLE 69.—QUANTITIES OF AIR DESIRABLE FOR VENTILATION[1]

Schools:	
Classrooms	30 cu. ft. of air per person per minute
Assembly rooms	15 to 25 cu. ft. of air per person per minute
Gymnasiums	12 changes per hour
Picture-machine booth	30 changes per hour
Dining rooms	12 to 20 changes per hour
Kitchens	20 to 60 changes per hour
Wardrobes and lockers	5 to 10 changes per hour
Toilet, bath, etc	10 to 20 changes per hour
Theaters:	
Seating spaces	30 cu. ft. of air per person per minute
Toilets, etc	20 to 30 changes per hour
Hotels:	
Assembly rooms	30 cu. ft. of air per person per minute
Dining rooms	10 to 20 changes per hour
Kitchens and laundries	20 to 60 changes per hour
Toilets and service rooms	12 changes per hour
Hospitals:	
Wards	40 to 60 cu. ft. of air per person per minute
Dining rooms	12 changes per hour
Toilets and service rooms	12 to 30 changes per hour
Kitchens and laundries	20 to 60 changes per hour

[1] A.S.H. & V.E. "Guide," 1929.

248. Air Distribution.—Merely to supply enough air to a room is not sufficient for good ventilation; it must be distributed in a fairly uniform manner so that each occupant receives approximately the specified amount.

The amount of **space allowed per occupant** is another important factor in ventilation. When a room is overcrowded it is difficult to move a quantity of air through it sufficient to maintain uniformly the proper effective temperature without causing uncomfortable drafts. Also, ample room volume is desirable as a reservoir of air to dilute the products of respiration. In schools a minimum of 180 cu. ft. per person is desirable; and in theaters, churches, and auditoriums not less than 90 cu. ft.

249. Ozone.—The use of ozone for the treatment of odors has been recurrently advocated for many years. Ozone generators, in which ozone is produced by an electrical discharge, are on the market and are designed to be used in connection with air-conditioning systems.

The neutralization of odors by ozone is brought about principally by the masking effect of the odor of the ozone itself, which is pleasing to some people but not to all. The highest *permissible* concentration in ventilating work is 0.1 part per million parts of air, and the quantities of odoriferous substances are usually of such a small order of magnitude that only about 0.01 part per million is needed. Greater concentrations are rather harmful, and quantities of the order of 15 parts per million are exceedingly irritating to the respiratory membranes and the eyes.

The use of ozone is not at present universally accepted as a desirable adjunct to an air-conditioning system, although it has some advocates.

With industrial odors the reaction of ozone with the odorous gasses or fumes may result in products that are decidedly harmful, and in a few cases really deadly gases may be produced. Ozone should therefore never be used in the ventilation of industrial buildings without knowledge of the effect of the oxidation of any gases that may be present.[1]

250. Ionization.—The effect of ions in the atmosphere on comfort, health, and working efficiency has received considerable attention. A feeling persists that there is some vitalizing property in open country air that has not yet been identified and that it may be related to ionization.

Yaglou, Benjamin, and Choate[2] found that the *small-ion* content in a room decreased from about 300 to about 75 per cubic centimeter when the room was occupied by a number of people. To maintain a high ionic content required a prohibitively high rate of air renewal.

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[1] For further discussion see F. E. Hartman, "Ozone and Its Use in Ventilation," *Trans. A.S.H. & V.E.*, 1924.

[2] See "Changes in Ionic Content of Air in Occupied Rooms Ventilated by Natural and by Mechanical Methods," *Trans. A.S.H. & V.E.*, 1932.

The ion content of air naturally increases with the temperature, and in cold weather the ion content indoors may therefore be higher than outdoors.

It is possible to create ions artificially, and there are machines offered for this purpose for use in ventilating work. Until a great amount of further work is done, however, on the physiological effects of ionization, and until its value has been conclusively demonstrated, it cannot be said to be of much practical significance in air conditioning.

251. Summary.—Satisfactory air conditioning, from the standpoint of human comfort and health, thus depends upon a number of factors, which may be listed as follows:

Air supply and distribution	This is the basis of ventilation.
Temperature	These are interrelated, and their combined effect is expressed by the **effective temperature.**
Humidity	
Air motion	
Radiation	May be an important factor under some conditions.
Odors	These factors may be said to determine the **purity** of the air.
Dust	
Bacteria	
Injurious substances	

No complete study of the conditions in an existing room can overlook any of these factors, and no system of air conditioning should be designed without considering them.

This chapter has dealt with the several factors that enter the problem of air conditioning and with the physiological reactions of the human body to its atmospheric environment. It has been seen that the problem of creating optimum conditions for comfort is very largely thermal and that it is by no means a simple one.

It may be said that only fairly complete knowledge now exists as to the conditions that should be maintained, and there is room for much improvement in the mechanical methods of creating such conditions. One difficulty has been the reluctance on the part of the general public to pay the price for the rather elaborate equipment involved. Another drawback has been the inexpert operating attention given to many installations. Both of these conditions are gradually being improved.

The present interest in summer cooling illustrates the increasing attention being given to the subject. This phase of air conditioning owes its development to the demonstration of the commercial value, in stores, theaters, restaurants, etc., of maintaining comfortable conditions. The further spread of summer cooling to residences, hotels, and offices will depend upon the value that can be demonstrated for it in those applications, and with the general trend toward improved standards it is probable that the future will see a steady enlargement of the field.

As to winter air conditioning, it should be evident from the material in this chapter that proper control of dry-bulb temperature is the first requirement; that humidification, filtering, and circulation of air are decidedly secondary; and that air conditioning to be properly accomplished requires apparatus and operating technique of the highest caliber.

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See also footnotes throughout the chapter.

Problems

1. Find the effective temperature for the following conditions:

Condition	Dry-bulb temperature, °F.	Wet-bulb temperature, °F.	Air velocity, feet per minute
a	70	60	300
b	72	67	0
c	68	61	100
d	75	65	500
e	100	85	0
f	80	74	300

2. In an auditorium occupied by 1,000 people with the dry-bulb temperature maintained at 75° and the wet-bulb at 60°, how much sensible heat, latent heat, and moisture will be given off by the audience per hour?

3. If the people in Prob. 2 were doing work at the rate of 16,538 ft.-lb. per hr. in the same atmosphere, what would be the sensible heat, latent heat, and moisture dissipation per hour?

4. Calculate Probs. 2 and 3 for conditions of 72° dry bulb and 65° wet bulb, which give the same effective temperature as in the previous cases, and note the difference in the results.

5. The dew point in a room varies from 60° to 47°. There is a negligible air velocity. If it were desired to maintain an exactly constant effective temperature of 71°, what would be the dry-bulb temperatures required at the two extremes?

6. If people are comfortable in a room in winter with a temperature of 75° and a relative humidity of 15 per cent, how much might the temperature be lowered if a humidifier were installed to raise the relative humidity to 30 per cent?

7. A residence has a volume of 10,200 cu. ft. and an air-charge rate of 1 per hour (70° basis). The heat loss per hour per degree difference in temperature between inside and outside is 2230 B.t.u. On a day when the average outside temperature is zero, the inside humidity is 15 per cent and the dry-bulb temperature is 74°. If a humidifier were added and equal conditions of comfort were maintained with a humidity of 30 per cent, would more or less heat be required, and how much? (The heat required to evaporate the water must of course be included.)

CHAPTER XV

VENTILATING AND AIR-CONDITIONING SYSTEMS

252. General.—The purpose of this chapter is to describe the types and the general arrangement of systems used for the ventilating and air conditioning of different kinds of buildings.

Although summer air conditioning has made the greatest impression on the public mind, there is a large field for winter air conditioning and for systems that give ventilation only, without much, if any, treatment of the air. Ventilating and air-conditioning systems are naturally divided into three classes:

1. **Ventilating systems.** The present definition of ventilation, as distinguished from air conditioning, is not very exact, but ventilation is generally taken to mean the circulation of air without much, if any, treatment other than heating it to the proper temperature.
2. **Winter air conditioning.** This includes the cleaning, heating, humidifying, and circulating of air.
3. **Summer air conditioning.** This includes cleaning, cooling, dehumidifying, and circulating.
4. **Complete air conditioning.** This includes both the winter and the summer functions.

Ventilation is used where the refinements and the cost of air conditioning in its more complete forms are not believed to be justified. The circulation of untreated or partially treated air accomplishes some of the objectives of air conditioning, such as the reduction of odors, but cannot control the effective temperature except through a limited range of conditions.

Winter air conditioning, since it does not require costly cooling or dehumidifying apparatus, is used in office buildings, churches, some auditoriums, schools, and residences where the owners do not feel that summer cooling is justified. Summer air conditioning alone is used in offices, small stores, and other places where the cooling effect is primarily sought. Such buildings often depend upon window ventilation in all other seasons. Complete air conditioning is used in theaters, some office buildings, large department stores, and other buildings where both the summer and winter features are desired. The climatic conditions, of course, influence the extent to which each kind of air conditioning is found necessary.

253. Natural Ventilation.—The ventilation of certain types of buildings can be accomplished fairly well by making use of the wind and of the temperature differences between the inside and outside.[1] Industrial buildings are often designed with no other ventilating systems, the air entering through open windows and leaving through skylights, roof ventilators, or other open windows.

The wind can be utilized to move large volumes through the building in summer, when the temperature difference is small and of little effect in ventilation. In winter, on the other hand, the temperature difference can be made use of, whereas the wind effect would cause excessive ventilation.

In designing for natural ventilation the windows should be arranged with approximately the same amount of opening on all sides of the building, and convenient operating mechanisms should be provided for the adjustment of the window openings.

Ventilation in summer by wind effect is for the most part a horizontal air flow, but ventilation through temperature difference is largely a vertical movement. In the latter case, there must be openings both at the top of the building, such as roof ventilators or monitors for the escape of air, and other openings near the ground for the ingress of air. Since the flow is inward near the ground and outward near the roof, it follows that there is an intermediate neutral zone where the tendency for the air to flow either in or out (because of temperature difference) is zero,[2] and openings near this intermediate level are much less effective than openings at the roof and ground levels. The amount of air that will flow through the building because of the temperature difference depends also upon the difference in elevation of the inlets and outlets and upon the amount of opening.

254. Roof Ventilators.—Various forms of roof ventilators as shown in Fig. 204 are available. They are so constructed as to be storm-tight and to permit the outflow of air regardless of the direction of the wind, and some of them are built so that the wind actually increases the flow by an aspirating effect, as at *c* (Fig. 204).

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[1] W. C. RANDALL, "Aeration of Industrial Buildings," *Trans. A.S.H. & V.E.*, 1928.

[2] J. E. EMSWILER, "The Neutral Zone in Ventilation," *Trans. A.S.H. & V.E.*, 1926.

Roof ventilators are of real value in providing a weatherproof exit for the air. They do not provide a positive flow, however, but are dependent upon the effect of the temperature difference and upon the aspirating effect of the wind.[1]

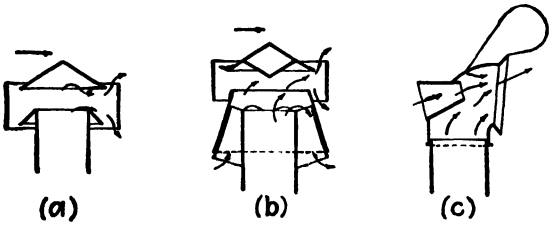

FIG. 204.—Roof ventilators.

255. Mechanical Systems.—The following discussion will consider the various mechanical systems for introducing and distributing air. The student should not be too greatly concerned whether a given system provides air conditioning or ventilation; the distinction is not a rigid one.

Mechanical systems are of two broad types: the central system, in which the air is treated at a central point and distributed through ducts, and the unit system, which consists of self-contained fan units located at suitable points throughout the building and usually having no ductwork. Mechanical systems may be classified as follows:

A. Central supply systems:

1. Split system—heating by radiators, fan used for ventilation only.
2. Heating divided—part of heat supplied by fan system, remainder by radiators.
3. Heating and ventilation by fan system.

B. Unit supply systems.

256. The Split System.—In the majority of systems the heating of the building is by radiators, and the fan system supplies air at nearly room temperature. With such an arrangement the fan system can be made use of to whatever degree is desired, and it is therefore often preferred for buildings like schools, which require ventilation only part of the time. The building can be brought up to temperature by the radiators, the fans being started when the occupants arrive.

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[1] F. B. ROWLEY, "Comparative Tests of Roof Ventilators," *Trans. A.S.H. & V.E.*, 1923; see also *Kansas State Agr. Coll. Eng. Exp. Sta. Bull.* 14.

If artificial cooling is used, it is supplied by the fan system. In fact, artificial cooling always involves the use of fans. There is no cooling unit that operates merely by radiation and natural convection to cool a room in the way a radiator warms a room.

257. Ventilation Combined with Heating.—In the opinion of many engineers it is advantageous to have the fan system supply part of the heat loss (about one-third) and to install radiators of a size sufficient to care for the remainder. The air is introduced at somewhat above the room temperature, so that there is less likelihood of uncomfortable drafts. Also, the room temperature is likely to be better regulated, because thermostats controlling the temperature of air in ducts usually work more accurately than those controlling radiators.

Still another method is to omit the radiators altogether and supply all the heat from the fan system. The temperature of the air supplied to the rooms is then determined entirely by the heat requirements. A system of this sort is fairly satisfactory; but in buildings where the occupants must be located near the windows, as in schools and office buildings, it is quite necessary to counteract in some other manner the cold downdraft at the windows if no radiators are installed. This downdraft, caused by the inleakage of air and the chilling effect of the cold glass, is always a factor to be reckoned with.

Sometimes the fan system is installed for heating purposes only, ventilation not being desired, and no fresh air is drawn from outside. The fan system in this case is designed to handle only the amount of air needed to convey the necessary heat. This is sometimes termed the "hot-blast" system. Factory buildings and other large spaces not densely occupied may be heated in this way, although unit heaters have largely superseded the central fan system in factory buildings.

258. Unit Systems.—The high cost of distribution ducts and the large amount of building space occupied by them led to the development of self-contained units placed in the individual rooms, sometimes two or more to a room.

There are many kinds of unit systems. Some units, for industrial buildings, merely heat the air and circulate it. Others are equipped with outdoor air connections and may also filter and humidify the air. There are also units for cooling only or for both heating and cooling. Units for heating and ventilating were discussed in Chap. V, and air-conditioning units are considered in detail in Chap. XXII. The development of unit heaters and air conditioners has been quite rapid, and they are now fairly satisfactory, although further improvement in such matters as temperature control is needed. For artificial cooling they are in many cases the most practicable system.

259. Air Conditioning of Office Buildings.—The use of air conditioning in office buildings has not become at all standardized, and it is possible to state only the present trends. In downtown office buildings, built for rental, complete systems of air conditioning are seldom installed throughout the office space because it has not yet been shown that the average tenant is willing to pay for it in the form of higher rental charges. However, separate fan systems are often installed for local areas at the desire of the tenant. They may be small central fan systems supplying the various rooms through ducts, or units for each room may be used. In either case the air conditioning is separated from the heating system, and the heating is usually done with ordinary radiators.

Figure 205 shows such a system that takes care of a small suite of offices in a large building. It has its own refrigerating unit and is self-contained.

There are, however, many office buildings built for and occupied by a single organization, and in such buildings air conditioning is often installed throughout. Large groups of people are employed in them, and air conditioning is needed in winter when window ventilation is unsatisfactory. Summer air conditioning is also frequently installed to improve working conditions and employee efficiency.

The split system (air conditioning separated from heating) is most frequently employed. Figure 206 shows a basement plan of an office building having a system for winter air conditioning only. It is a trunk duct system supplying air at approximately room temperature. Branches and risers convey the air to various parts of the building. Such a system would not be very suitable for summer air conditioning because it does not lend itself to temperature control for the various portions of the building, whose cooling requirements vary widely throughout the day because of the movement of the sun.

An important installation of complete air conditioning in an office building is that of the Metropolitan Life Building in New York City. It is illustrated in Figs. 207, 208, and 209. Figure 207 shows the elevation (somewhat diagrammatically). The fresh air is taken from outside at the fifteenth floor, where it is warmed or cooled, according to the season, and conveyed upward and downward to the other floors. At each floor are fans that mix the fresh air with a larger quantity of recirculating air and distribute the mixture locally. Cooling and filtering apparatus are provided for each fan unit.

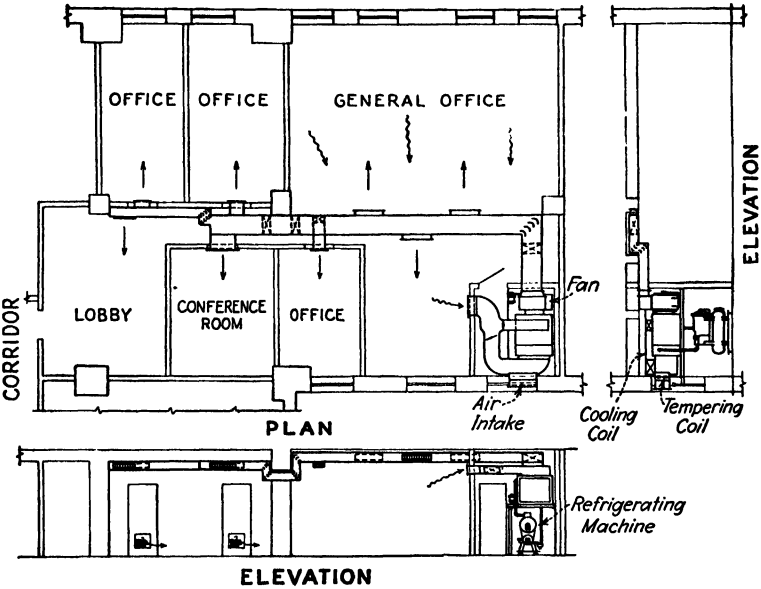

FIG. 205.—Air-conditioning system for an office suite.

Figure 208 shows the distribution ducts of a typical floor. The air is introduced into the rooms horizontally through grilles near the ceiling, which is stepped at the grille faces so that its height is increased toward the outer wall. This feature is unique.

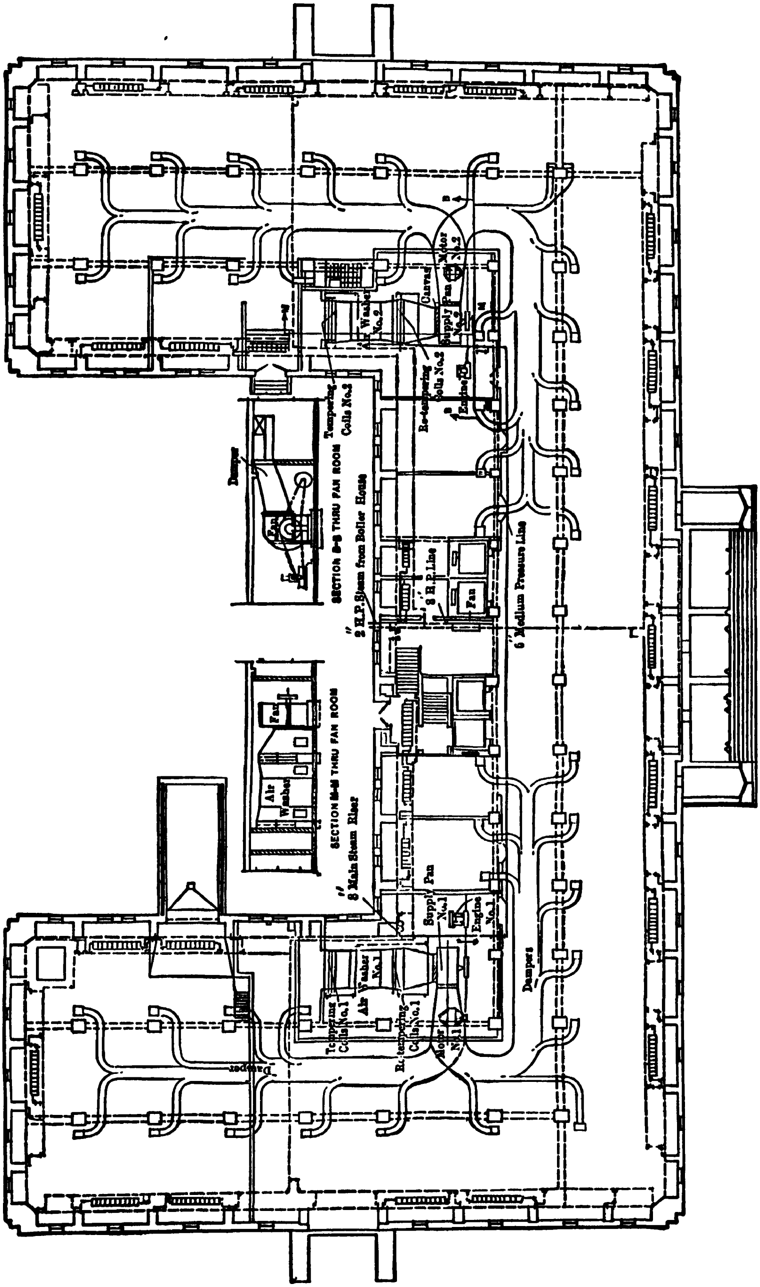

FIG. 206.—Basement plan of office building showing trunk ducts of ventilating system. (*Ammerman & McColl, Consulting Engineers.*)

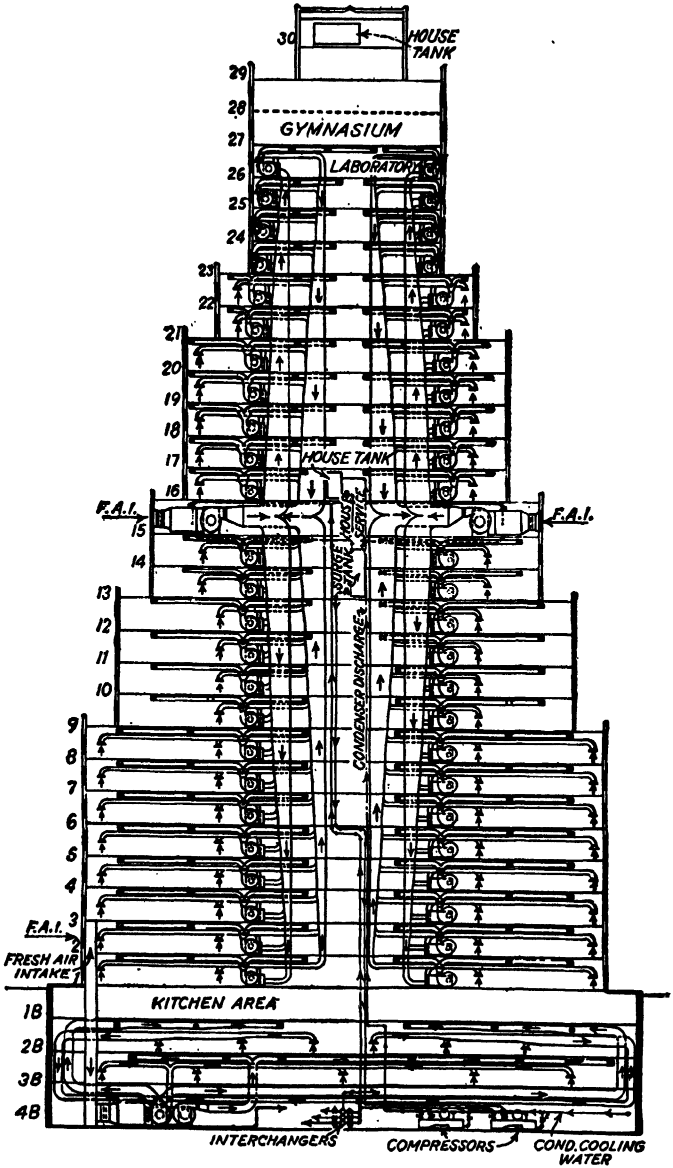

FIG. 207.—Elevation diagram—Metropolitan Life building. (*From E. C. Parker, "Air Conditioning and Ventilating a 31-story Office Building," Heating and Ventilating, August,* 1933.)

Figure 209 shows the exhaust system. It is necessary to remove and discharge to the outdoors a quantity of air equal to the amount of fresh air introduced. This quantity can largely be made up of exhaust from toilet rooms, restaurants, etc., which should not be recirculated.

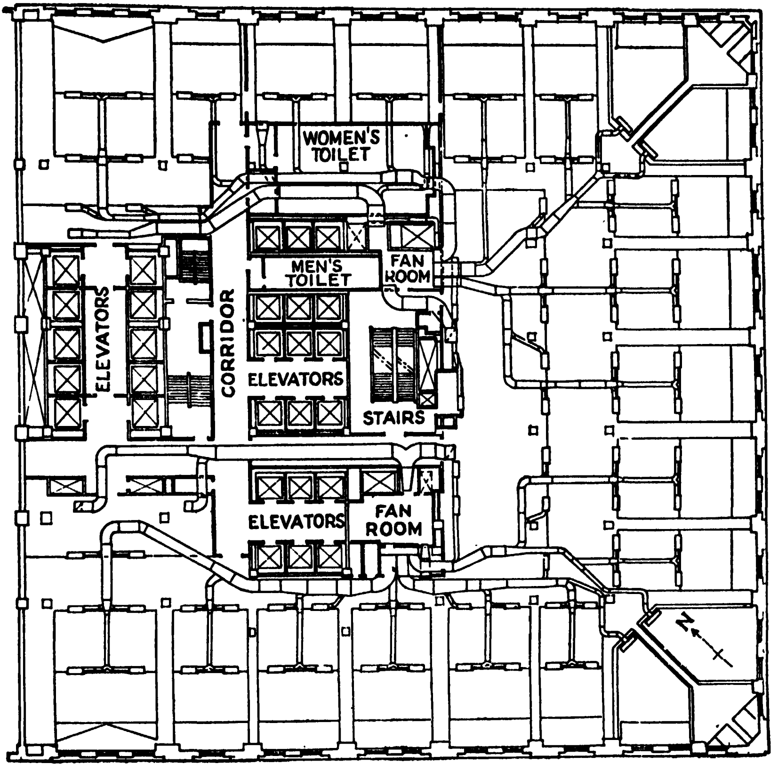

FIG. 208. Plan of typical floor—Metropolitan Life building. (*From McConnell and Kagey, "The Air-conditioning System of the New Metropolitan Building—First Summer's Experience," Trans. A.S.H. & V.E.*, 1934.)

A new trend in office-building design is the construction of windowless buildings using glass brick or solid masonry walls.[1,2,3]

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[1] R. E. HATTIS, "Unique Air Conditioning System Serves New Industrial Building," *Heating, Piping and Air Conditioning*, February, 1937.

[2] "First Windowless Office Building (Hershey Chocolate Company, Hershey, Pa.)," *Refrig. Eng.*, February, 1936.

[3] J. H. WALKER, "Air Conditioning Influences Office Building Design," *Heating, Piping and Air Conditioning*, October, 1937.

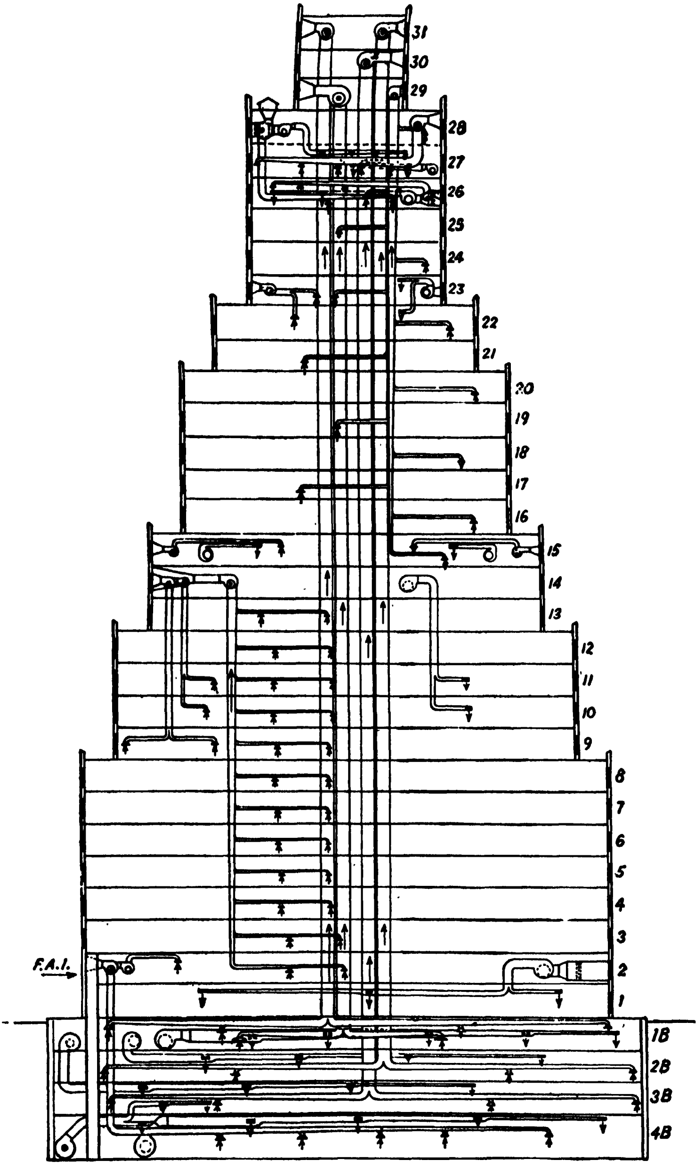

FIG. 209.—Exhaust system of Metropolitan Life building. (*From E. C. Parker, "Air Conditioning and Ventilating a 31-story Office Building," Heating and Ventilating, August,* 1933.)

In such buildings the air-conditioning system must be operated every working day of the year. There are no intermediate periods in spring or fall when neither heating nor cooling is needed and when open windows could be used. A similar condition arises when double windows are designed never to be opened for ventilation. The air-conditioning system for such buildings is basically the same as for the ordinary type, but it must be even more carefully designed, particularly for temperature control, and it must be extremely reliable, because the use of the building depends upon it.

260. School Ventilation.—Summer air conditioning is not needed or used in schools. School ventilation, however, is one of the largest fields for the ventilating engineer, and in modern schools the ventilating systems are often quite elaborate. A great deal of criticism has been directed against the methods used, and in fact there is much room for improvement in the operation, maintenance, and design of many school ventilating systems.

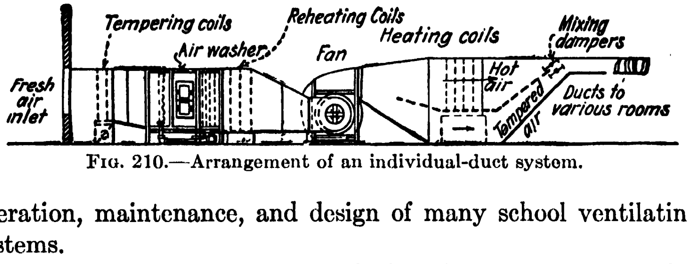

Fig. 210.—Arrangement of an individual-duct system.

The unit-ventilator system, in which each classroom is provided with one or more self-contained fan and heater units, has many advocates. Systems using a central fan will be considered in the following discussion.

One of the common methods of ventilation used in schools is the split system, in which the air for ventilation is introduced at a temperature of 63 to 72°, depending upon the outside temperature.

The individual-duct system supplies part or all of the heating requirements. Such a system is shown in Fig. 210. It is arranged to supply hot air (90 to 130°) and tempered air (68 to 70°), the two supplies being mixed by thermostatically controlled dampers and carried by individual ducts to the various rooms, each duct having its own mixing damper controlled from a thermostat located in the room itself. The temperature of the air in each duct is therefore just sufficient to supply the requisite amount of heat to the room.

The air upon entering from outdoors is passed through a tempering heater, which raises its temperature somewhat above the freezing point. It then flows through the air washer and a reheater and is drawn into the fan. The fan discharges it through an enlarging duct to the heating coils. Part of the air passes through the coils and is heated to from 90 to 130°, depending upon the outside temperature, and a portion passes below the heater and enters the tempered-air chamber at a temperature of about 68°.

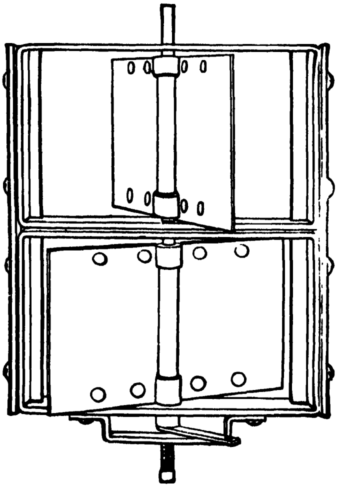

FIG. 211.—Mixing damper.

The mixing damper is shown in Fig. 211. The hot-air and tempered-air dampers are mechanically connected so that one closes as the other opens. The damper frames may be arranged in the same plane, as in Fig. 211, or at right angles, as in Fig. 210.

The hot- and the tempered-air chambers should be separated by a double decking, or they should be insulated from each other to prevent the transfer of heat. A basement plan of a school building with a system of this kind is shown in Fig. 212. The individual-duct system is satisfactory and there are many in use, but the cost of the sheet-metal ducts, as may be inferred from a study of Fig. 212, is very high, and this system has to some extent given way to other designs.

261. Plenum-chamber System.—A successful type of system for schools is that shown in Fig. 213. Modern schools are usually built without full basements, but with this type of system a plenum chamber of concrete or masonry extends beneath the corridor and carries air at approximately 68°. For each tier of rooms (a first-floor room and a second-floor room) there is a set of heating coils, and for each room there is a thermostatically controlled mixing damper, which takes part tempered and part

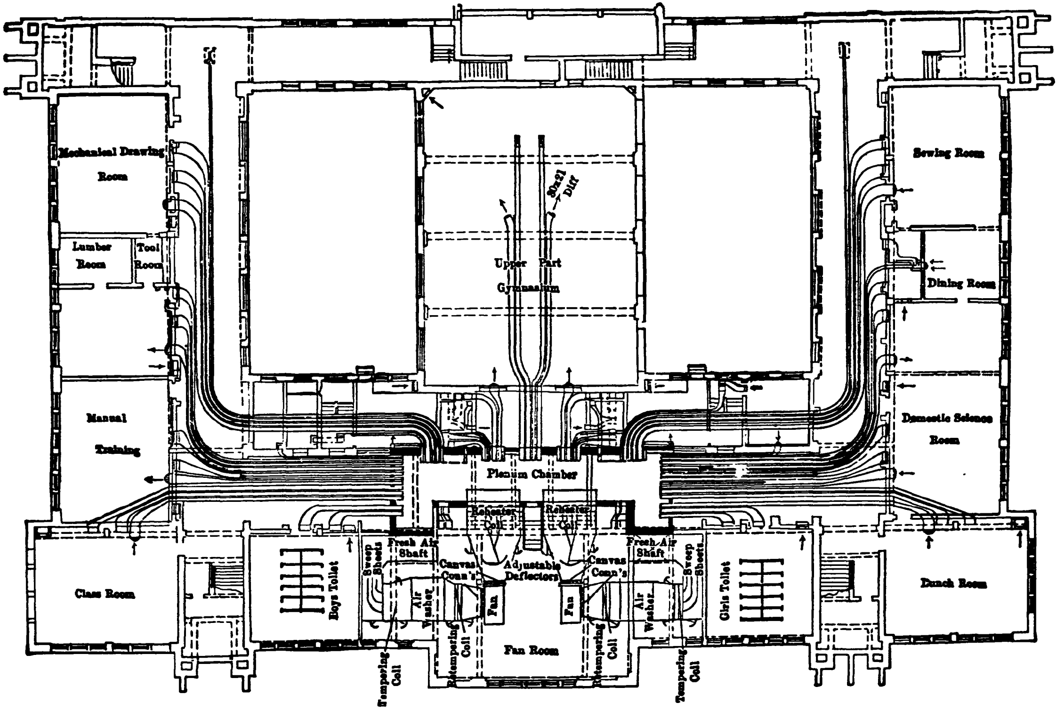

FIG. 212.—Basement plan of school building equipped with individual-duct system. (*Ammerman & McColl, Consulting Engineers.*)

heated air as required. The thermostat is located in the room itself. The supply and exhaust ducts are installed in the corridor partition, and the main exhaust duct is above the second-floor corridor.

It is evident that this is a simple and relatively cheap system to build. The large plenum chamber is readily cleaned—an important factor—and serves as a tunnel for various pipes.

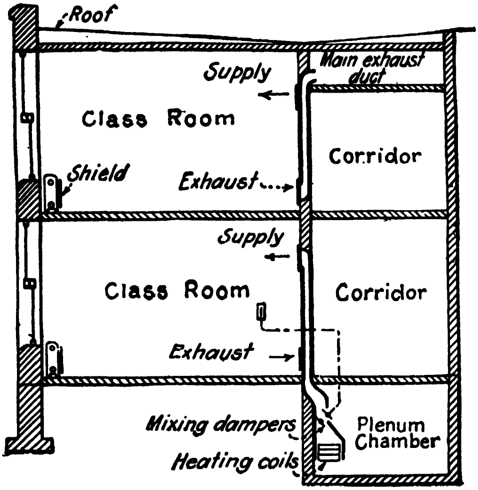

FIG. 213.—Plenum-chamber system.

262. Exhaust Ducts.—Provision must be made for removing the air from the rooms at the same rate at which it is supplied, and a system of vent flues is provided for that purpose. The flues from the separate rooms join in a trunk duct and lead to a common discharge at the roof. In schools the attic is sometimes used as a discharge chamber, the flues leading directly to it. Exhaust flues are figured at a velocity of 600 to 750 ft. per min. and are assumed to carry off the same amount of air as is delivered to the room. In some cases an exhaust fan is installed to facilitate the removal of the foul air. The velocity in the exhaust flues can then be from 1,200 to 1,500 ft. per min. In public buildings over three or four stories in height, where the friction in the exhaust flues is appreciable, an exhaust fan is desirable.

263. Recirculation.—If it is admitted that the principles governing air conditioning which have been outlined are correct, there seems to be little objection to recirculating the air in such buildings as schools if it is reconditioned and deodorized during the cycle. Not only is there a large fuel saving with recirculation, but the installation cost of the system can be considerably reduced, because less boiler and air-heater capacity are required.

The amount of permissible recirculation is usually limited by the building up of humidity in the rooms. The humidity that the air will pick up in an air washer naturally is greater when the incoming air is at a higher temperature; so that when recirculated air is used, the tendency is for the humidity to become excessive. The most satisfactory solution is to mix cold fresh air and recirculated air in such proportions that the air entering the washer is at a temperature of, say, 40 to 50°. The amount of recirculation is thus considerably greater in cold weather and is cut off completely in mild weather. If the air outside is at 0°, the recirculated air 65°, and the mixture 50°, it is evident that nearly 77 per cent of the total is recirculated air.

In many schools this method of recirculation is an established and apparently satisfactory practice. Odors do not build up objectionably (owing probably to the rewashing of the air and to the dilution effect), and there is no evidence that contagion is increased.

264. Factory Heating.—The hot-blast system is often the best system for industrial buildings, as it affords a means of supplying fresh air to replace that containing the fumes or moisture from manufacturing processes. It is also desirable in factory buildings where the space required by direct radiators cannot be spared. Owing to the fact that such buildings are seldom divided into many rooms, the air can be supplied at a constant temperature through a trunk system of ducts. A draw-through arrangement is almost universally used, the heating coils being placed on the suction side of the fan, which discharges directly into the main duct. For ordinary shop buildings of steel construction, the ducts are of galvanized iron and are suspended from the columns or roof trusses. An example of this arrangement is shown in Fig. 214.

The trunk-duct system has been largely superseded in many kinds of industrial buildings by the unit-heater system. The central-fan system is still sometimes used, however, where more than three or four units would be required.

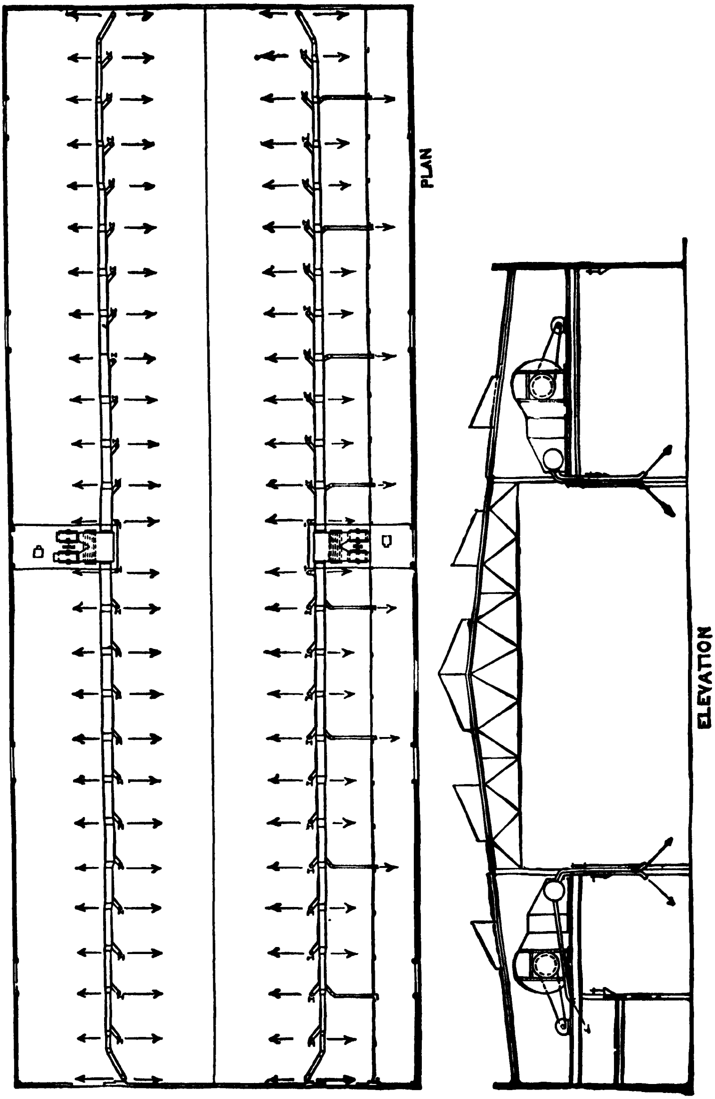

FIG. 214.—Hot-blast system in factory building.

One ever-present problem in factory heating is to produce the proper conditions in the occupied zone without overheating the upper portion of the room or building as a result of the natural tendency of the warm air to rise. In this respect both the central-fan and the unit systems are superior to direct radiators, as they tend to maintain better heat distribution. It is better to circulate a large quantity of air at a moderate temperature than a small quantity at a higher temperature. The maximum air temperature, carried only in the coldest weather, should not exceed 130°.

265. Fan Systems for Churches, Theaters, and Auditoriums.—Buildings of this class are usually both heated and ventilated by the fan system, except that where there are windows in the auditoriums, as in churches, it is advisable to install direct radiators under them to counteract the cold downdraft. The offices, entrance lobby, stage, etc., of such buildings require direct radiators. The air-conditioning requirements are paramount; in fact, after the audience has gathered, the problem is usually one of cooling rather than heating.

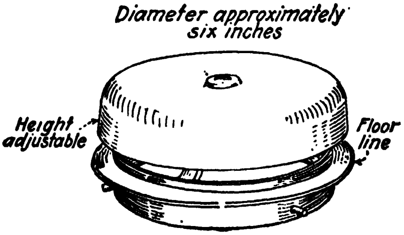

FIG. 215.—Mushroom ventilating head.

The method of air introduction will depend upon whether the room is fitted with permanently fixed seats or has movable seats and a floor that is sometimes cleared. In the former case good results can be secured by the use of the mushroom system of exhaust with exhaust outlets of the type shown in Fig. 215, spaced at regular intervals under the rows of seats. For spaces utilizing movable seat, best results are obtained by using exhaust grilles located on the walls near the floor line. Generally, the air is introduced from overhead through grilles designed to harmonize with the decorative scheme.

A theater system of modern design is shown in Fig. 216. The air is introduced at the ceiling and beneath the balcony and is removed through grilles at the rear of the main floor and from an exhaust chamber below the balcony seats. The mushroom system is often used instead of the large exhaust grilles on the main floor, but it costs somewhat more and therefore is not universal.

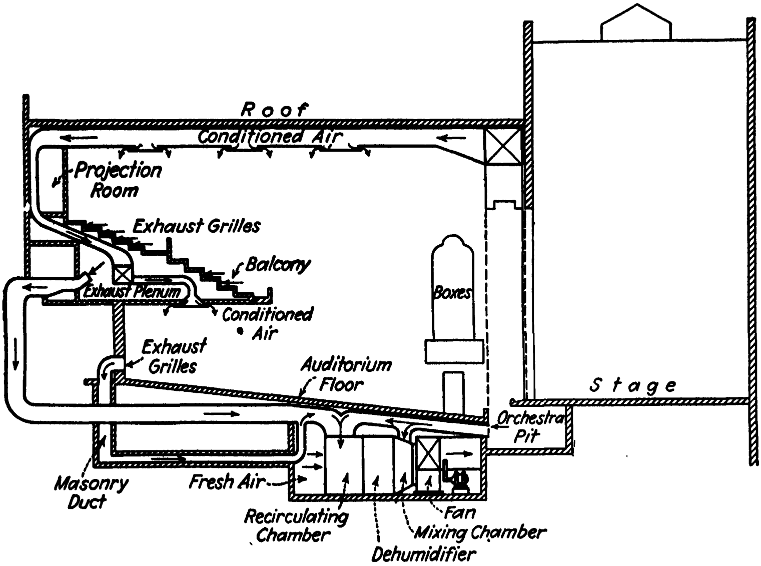

FIG. 216.—Typical theater installation.

266. Ventilation of Kitchens, Toilet Rooms, Etc.—There are certain kinds of rooms where copious ventilation is required because of the occurrence of odors, fumes, or injurious gases. The air from such rooms is not recirculated but is exhausted to the out-of-doors. When such conditions exist, it is important to have an exhaust system of ample capacity, and in some cases no system of air supply other than inleakage through cracks and open windows is necessary. In toilet rooms, for example, it is usually quite satisfactory to provide only an exhaust system and permit windows to be left partly open.

The quantities of air required for satisfactory ventilation of this kind are greater than is usually appreciated. Air renewal at the rate of 20 changes per hour is the minimum for kitchens and public toilet rooms, and a rate of 30 changes per hour is preferable for the latter.

Where the inside air is very moist, it is not satisfactory to draw in cold air because of condensation as it encounters the warm, moist air. The air supply must be warmed or drawn from other parts of the building.

Where there are vapors or fumes, exhaust outlets are usually placed near the ceiling of the room, unless the vapors are known to be heavier than the atmosphere.

267. Systems for Other Types of Buildings.—Other kinds of buildings in which systems of air conditioning are used include stores, hotels, and residences. In stores ventilation is required for basement space, and summer cooling has become important for basement and other floors. The split system is often used, and a satisfactory job can be done with self-contained units.

The lobbies and restaurants of hotels are usually air-conditioned, and here also the split system is popular, although the use of the fan system for heating is not uncommon.

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"Comparative Analysis of Office Building Air Conditioning Systems," by J. R. Hertzler, *Heating, Piping and Air Conditioning*, May, 1938.

"Double Duct Air Conditioning System Meets Individual Room Requirements," by J. N. Livermore, *Heating, Piping and Air Conditioning*, February, 1942.

The following magazines in various issues during recent years contain descriptions of important air-conditioning installations:

Heating and Ventilating.

Heating, Piping and Air Conditioning.

CHAPTER XVI

FANS, FILTERS, WASHERS, AND COILS

268. Flow of Air in Ducts.—When air, like other fluids, is moved through a pipe or duct, a certain pressure or head is necessary to start and maintain the flow. This head has two components. The **static head** is that which is required to overcome the frictional resistance of the air against the surface of the duct. The **velocity head** is the pressure required to produce the velocity of flow. The sum of these two components is termed the **total head.**

The static and velocity heads are mutually convertible. The velocity head is a function of the velocity of flow; and if the velocity in the duct is decreased at any point because of an increase in the cross-sectional area, a portion of the velocity head will be converted into static head. Conversely, when the area is reduced, the static head is partially converted into velocity head. The interchange, however, is always accompanied by a certain amount of net loss of head, depending upon the abruptness of the change in area and shape of the section in which the change of area takes place. To conserve total head (and therefore power) it is necessary to make any necessary changes in the cross section of air ducts in a gradual manner.

The velocity head may be considered as the height of a column of air that will have at its base a pressure sufficient to produce the given velocity, the relation being represented by the common expression $v^2 = 2gh$. To express the velocity head in inches of water, the usual standard, let

D = density of air under the given conditions, lb. per cu. ft.
D^1 = density of water = 62.3 lb. per cu. ft. at 70°.
h_v = velocity head, in. of water.
h = velocity head, ft. of air.

Then

$$hD = \frac{h_v}{12} D' \qquad \text{or} \qquad h = \frac{h_v}{12}\frac{D'}{D}$$

$$V^2 = 3{,}600 \times 2g \times \frac{h_v D'}{12D}$$

where V is the velocity, ft. per min.

$$V = 1{,}096.5\sqrt{\frac{h_v}{D}} \tag{1}$$

$$h_v = \left(\frac{V}{1{,}096.5}\right)^2 D \tag{2}$$

For standard air (70° and 29.92 in. barometer)

$$V = 4{,}005\sqrt{h_v} \tag{3}$$

The static head or pressure in an air duct may be thought of as the pressure tending to burst the duct. It may be readily measured by means of a water gage communicating with the duct in the manner shown at A in Fig. 217. The deflection of the water levels indicates the static pressure in inches of water. The total or dynamic head is measured by a tube whose open end points against the flow as at B. Since the velocity varies at different points in the cross section of the duct, any single reading of the total pressure applies only to the particular location of the tube in the duct. The velocity head, which is equal to the difference between the total and static heads, can be computed from them or can be measured directly by connecting the U tube as at C in Fig. 217.

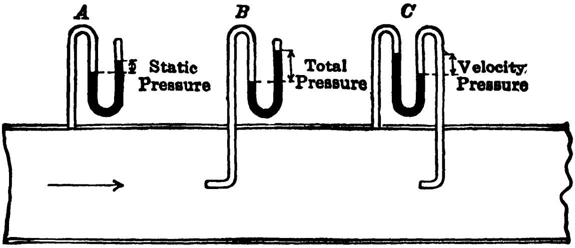

FIG. 217.

269. The Pitot Tube.—The relation between the velocity and the velocity head affords a convenient method for measuring the flow of air through pipes using the pitot tube, illustrated in Fig. 218. The tube is inserted into the pipe in such a manner that the head AB is parallel to the flow of air, with the end A toward the flow. The part AB consists of an inner tube, which transmits the total pressure to the tube C, and an outer jacket through which the static pressure is transmitted to the tube D.

This outer jacket contains several small holes through which the static pressure is transmitted. The two pressures are transmitted to the ends of the slant gage E, which is a U tube arranged with one leg at an angle so that the linear deflection per inch of height is increased. Gages of this type are usually filled with oil but are calibrated to read in inches of water column. The reading on the gage is evidently the velocity head, as it is the difference between the total and static heads.

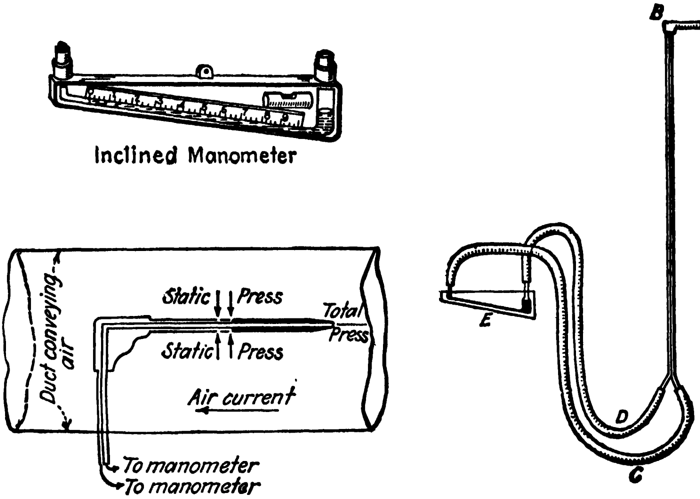

FIG. 218.—Pitot tube.[1]

The velocity of flow is not the same at all points in the cross section of the duct. Near the walls it is retarded by friction, and it reaches a maximum at the center of the pipe. It is therefore necessary to measure the velocity at several points in the pipe in order to obtain an average figure.

In a square or rectangular duct the cross section is divided into several equal rectangles, and readings are taken with the pitot tube at the center of each of these divisions. The velocity corresponding to the pressure at the point where each reading is taken is then computed from Eq. (1), Art. 268, in feet per minute. The average of these computed velocities is taken as the average velocity in the pipe. The quantity of air flowing can be readily computed from the velocity and the cross-sectional area of the pipe.

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[1] For exact details see Standard Code for the Testing of Centrifugal and Axial Fans, 3d ed., 1938. *Nat. Assoc. Fan Manufacturers Bull.* 103.

For a round pipe the cross-sectional area should be divided into a number of annular zones of *equal area,* and a traverse of the pipe should be made in both a vertical and a horizontal direction, as shown in Fig. 219. For each foot of pipe diameter the cross section should be divided into at least three of these zones. Table 70 gives the distance from the center of the pipe at which each reading should be taken in per cent of the pipe diameter.

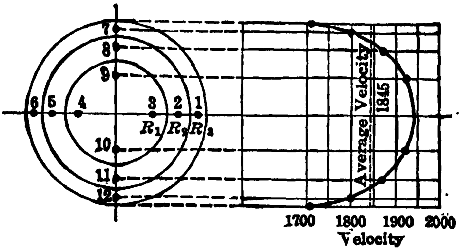

FIG. 219.—Division of round pipe into annular zones.

TABLE 70.—PIPE TRAVERSE FOR PITOT-TUBE READINGS
(Distance from center of pipe to point of reading in per cent of pipe diameter)

No. of equal areas in traverse	No. of readings	1st R_1	2d R_2	3d R_3	4th R_4	5th R_5	6th R_6	7th R_7	8th R_8
3	12	20.4	35.3	45.5					
4	16	17.7	30.5	39.4	46.6				
5	20	15.5	27.2	35.3	41.7	47.4			
6	24	14.5	25.0	32.3	38.2	43.3	47.9		
7	28	13.4	23.1	29.9	35.3	40.1	44.3	48.2	
8	32	12.5	21.6	28.0	33.2	37.6	41.5	45.1	48.4

It is important that the velocities be computed separately and averaged, for the velocity varies as the square root of the pressure, and accurate results cannot be obtained by averaging the pressure readings. The method outlined above is part of the fan-test Code.

270. Theory of the Centrifugal Fan.—The centrifugal fan consists fundamentally of a wheel having several radial vanes revolving in a casing. Air enters near the axis of the wheel, flows to the circumference under the influence of the centrifugal force produced by the rotation, and is discharged through the outlet, which is located tangentially with respect to the fan wheel.

The flow of air through the fan is caused by (*a*) the centrifugal force imparted to the air enclosed between the vanes and (*b*) the tangential velocity imparted to the air as it leaves the tips of the blades. To convert this velocity into useful static pressure, the fan housing is given a scroll shape of gradually increasing area; and further reduction of velocity is desired, an enlarging discharge piece is added to the fan outlet.

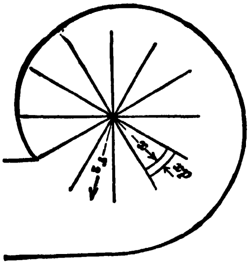

FIG. 220.

Figure 220 represents an elementary centrifugal fan. Consider a thin layer of air of thickness dx between two of the vanes at a distance x from the axis and having an area of S. The volume of this layer of air is then Sdx; and if its density is D, then the weight will be $SdxD$. Assume that the fan outlet is completely closed and that the wheel revolves at the rate of ω radians per second. Then the centrifugal force[1]

$$df = \frac{\omega^2 SdxD}{g}$$

The unit pressure dp coresponding to df is evidently $= df/S$, and the equivalent head

$$dh = \frac{dp}{D} = \frac{df}{SD}$$

Then

$$dh = \frac{\omega^2 xdx}{g}$$

Let r_1 be the radius at the base of the blade and r_2 the radius at the tip. Then

$$h = \int_{r_1}^{r_1} \frac{\omega^2 xdx}{g} = \frac{\omega^2 r_2^2 - \omega^2 r_1^2}{2g}$$

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[1] Centrifugal force $= w\omega^2 r/g$ for a weight w at radius r.

If the entire column of air between the two blades from the axis to the radius r_2 is assumed to be affected, then $r_1 = 0$, and

$$h = \frac{\omega^2 r_2^2}{2g}$$

If v is the linear tip speed, then $v = \omega r_2$, and

$$h = \frac{v^2}{2g}$$

The second source of pressure is the equivalent of the velocity v of the air at the blade tips, which is equal to

$$h' = \frac{v^2}{2g}$$

The total pressure or head developed under the assumed conditions would then be that equivalent to the resultant of the two velocities, as indicated in Fig. 221.

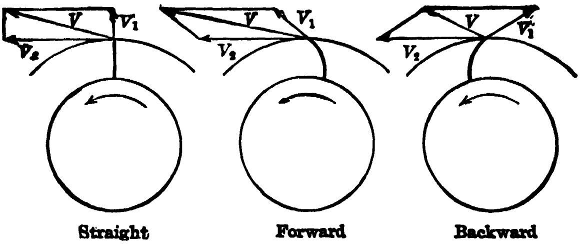

FIG. 221.—Effect of various blade designs.

The foregoing analysis is approximate only and is complicated under actual conditions by the effect of the various sources of pressure loss and by the fact that the conversion of the velocity head into static head is only partial. The analysis serves to show, however, the relation between the pressure developed by a centrifugal fan and the fan speed.

271. Fan Blades and Housings.—Fan blades may be designed in any one of three ways: radial, curved forward (that is, in the direction of rotation), or curved backward. In Fig. 221 is shown graphically the effect of the different blade designs on the resultant velocity of the air. The air leaving the tip of the blade has a velocity component v_1, parallel to the blade, and a tangential component v_2. If the blade is curved forward, the resultant velocity v will be greater for the same peripheral speed than that in the straight-blade type, and if it is curved backward, the resultant velocity will be decreased. For a given pressure the fan with backward-bent blades requires a higher rotative speed than the other types and is therefore in some cases better adapted to direct motor drive.

The velocity head developed by the fan wheel is considerably greater than is required, whereas the static head, which is the force necessary to move the air against the frictional resistance of the duct system, is less than is required. The velocity head is therefore partially converted into static head by designing the housing in a suitable scroll shape so that the velocity of the air is gradually reduced. The efficiency with which the conversion to static head takes place depends upon the proper design of the housing. It is the static head developed by a fan that is useful in overcoming duct resistance; and before the velocity head can become available, it must be converted into static head. Generally speaking, the fan that has the greater static head in proportion to velocity head is the more desirable.

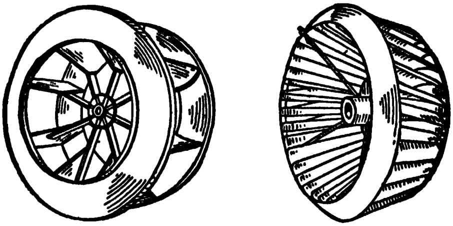

FIG. 222.—Wheels of steel-plate and multiblade fans.

272. Commercial Forms of Fans.—Centrifugal fans are of two general types. The older type is the so-called "steel-plate" fan (although strictly speaking nearly all fans are built of steel plate) having a relatively small number of nearly flat radial blades. The multiblade fan, a somewhat more recent development, has a large number of short, curved blades set near the periphery of the wheel. The steel-plate fan is not usually so efficient as the multiblade fan, although it can be made so if the blades are very narrow and of large diameter. The multiblade fan occupies less space and operates at a higher speed and is therefore more suitable for direct connection to motors. In general, the multiblade fan is standard for ventilating work. The two forms are shown in Fig. 222, and Fig. 223 shows a multiblade fan completely assembled.

The multiblade fan is built with forward-curved blades, backward-curved blades, and blades of combined curvatures. All

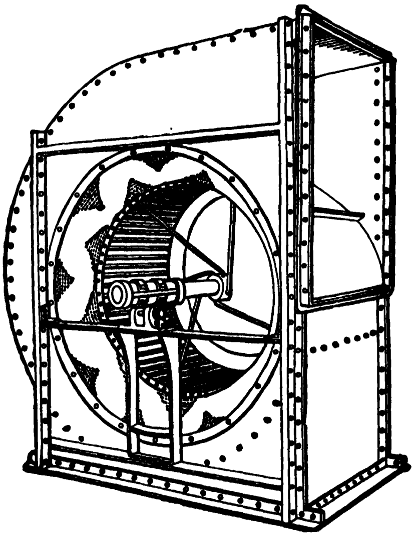

Fig. 223.—Assembly of multiblade fan.

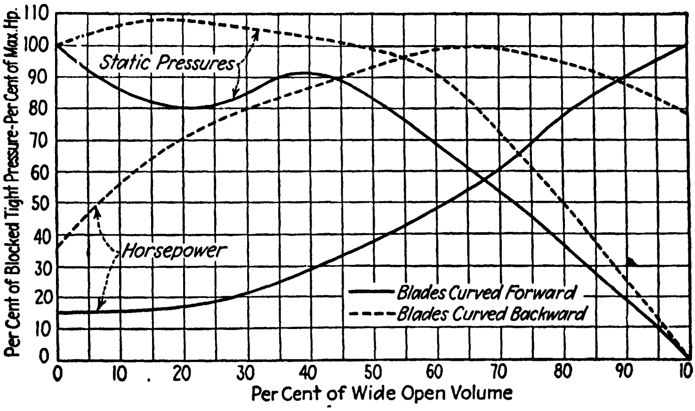

Fig. 224.—Characteristics of multiblade fans.

types are used in air-conditioning work, but each has certain limitations.

The important difference in the characteristics of forward- and backward-curved blade fans is shown by the curves of Fig.

224. The static-pressure curve for the forward-curved blade has a hump in it, and the horsepower curve rises continually and is higher at the wide-open volume than in the operating range. It is therefore possible to overload the motor, though there is little danger if a motor with a reasonable margin of capacity is selected.

The static pressure curve for the backward-curved blade droops continuously from a point of small volume, and the horsepower curve droops at the larger volumes.

Because of its higher speed, the backward-curved blade fan must be more ruggedly built. It may often be direct-connected, a feature that sometimes conserves space, although this feature is not so important since the general adoption of short V belts in belt drives.

A fan sufficiently large to operate near its point of maximum efficiency should be selected. An overloaded fan is much more apt to be noisy.

273. Air Horsepower and Input Horsepower.—The power required for moving air through a system of ducts may be expressed as follows:

Let

p = total pressure, in. of water.
a = cross-sectional area of duct, sq. ft.
v = velocity of air, ft. per min.

Then, applying the necessary conversion factors to change the total pressure to pounds per square foot,

$$Ahp = \frac{pav \times 144}{12 \times 2.31 \times 33{,}000}$$

or

$$Ahp = 0.000157\ pav \tag{4}$$

If q is the volume of air delivered per minute in cubic feet, then $q = av$, and

$$Ahp = 0.000157pq = \frac{pq}{6{,}356} \tag{5}$$

where the pressure p is expressed in inches of water.

If the pressure is expressed in terms of the equivalent cloumn of air of height h, then

$$Ahp = \frac{hDQ}{33{,}000} \tag{6}$$

where D = the density of the air, lb. per cu. ft.
Q = the air volume, cu. ft. per min.

In a fan the actual head developed is only a portion of the theoretical head v^2/g and is represented approximately by kv^2/g. The input horsepower required to drive a fan is then

$$Hp = \frac{ckv^2}{g} \times \frac{DQ}{33{,}000} \tag{7}$$

where c is a factor that takes into account the mechanical losses in the fan. Combining all of the constant factors,

$$Hp = Kv^2QD \tag{8}$$

v being the peripheral velocity, which varies directly as the speed of the fan. Since Q varies directly as the speed, the power required varies as the cube of the speed.

274. Fan Efficiency.—The true, or **mechanical, efficiency** of a fan is the ratio of the horsepower output to the horsepower input at the fan shaft. The horsepower output is a function of the static pressure developed by the fan and of the velocity pressure. The velocity pressure, however, is not entirely convertible into useful work, because it cannot be entirely converted into static pressure. The extent to which it is converted depends somewhat upon the ductwork design. Therefore a method of calculating the efficiency based on the static pressure is often used. This **static efficiency** is the mechanical efficiency multiplied by the ratio of static pressure to total pressure.

The mechanical efficiency is expressed by the formula

$$ME = \frac{Ahp}{\text{input } hp} \tag{9}$$

$$ME = \frac{pq}{6{,}356\ hp} \text{ (From Eq. (5) Art. 273)} \tag{10}$$

where hp = input horsepower.

$$\text{Static efficiency } SE = \frac{pq}{6{,}356\ Hp} \times \frac{\text{static pressure}}{\text{total pressure}}$$

$$SE = \frac{psq}{6{,}356\ Hp} \tag{11}$$

where ps = static pressure.

275. Fan Performance.—Certain laws govern the performance of fans, and these are useful in selecting a fan and in determining the effect of changing the conditions of operation. From a consideration of Art. 270 the following three laws may be stated for a given fan size, piping system, and air density:

1. The capacity varies directly as the fan speed.
2. The static pressure varies as the square of the fan speed.
3. The horsepower varies as the cube of the fan speed.

Example.—A given fan delivering 10,000 c.f.m. at a static pressure of 1 in., speed 600 r.p.m., and drawing 4 hp., is required to deliver 16,000 c.f.m. What will be the speed, static pressure, and horsepower?

$$\text{Speed} = 600 \times \frac{16{,}000}{10{,}000} = 960 \text{ r.p.m.}$$

$$\text{Pressure} = 1 \times \left(\frac{960}{600}\right)^2 = 2.56 \text{ in.}$$

$$\text{Horsepower} = 4 \times \left(\frac{960}{600}\right)^3 = 16.38 \text{ hp.}$$

From the foregoing the following additional relations are obvious:

4. The capacity and the speed vary as the square root of the pressure.
5. The horsepower varies as the $(\text{pressure})^{3/2}$.

Example.—With the fan of the previous example, what would be the capacity, speed, and horsepower if the static pressure is changed from 1 in. to 1.75 in.?

$$\text{Capacity} = 10{,}000 \times \left(\frac{1.75}{1.00}\right)^{1/2} = 13{,}230$$

$$\text{Speed} = 600 \times \left(\frac{1.75}{1.00}\right)^{1/2} = 794$$

$$\text{Horsepower} = 4 \times \left(\frac{1.75}{1.00}\right)^{3/2} = 9.3$$

When the density of the air varies

At constant pressure

6. The speed, capacity, and horsepower vary inversely as the square root of the density.

At constant capacity and speed

7. Horsepower and pressure vary directly as the density.

For a constant weight of air

8. Capacity, speed, and pressure vary inversely as the density.
9. Horsepower varies inversely as the square of the density.

The density, of course, depends upon the pressure and temperature, varying directly as the barometric pressure and inversely as the absolute temperature.

Example.—In a given system the fan requires 9.5 hp. when the air temperature is 20°. How much more power will be required for the same weight of air if the temperature is increased to 110°?

$$9.5 \times \left(\frac{459.7 + 110}{459.7 + 20}\right)^2 = 13.4 \text{ hp.}$$

$$13.4 - 9.5 = 3.9 \text{ hp.}$$

276. Selection of a Fan.—The considerations involved in selecting a fan are

1. **Over-all dimensions,** which are often critical where space is limited.
2. **Tip speed,** which is an important criterion of quiet operation. In general, the lower the tip speed for a given type of fan, the less the noise.
3. **Quietness of operation,** which varies with different makes and types of fans. A noise rating can be submitted by most fan manufacturers and is often desirable.
4. **Efficiency,** which of course determines the power comsumption.
5. **Mechanical** construction features, such as strength of parts and thicknesses of plates.
6. **Cost.**

It is usually possible to obtain the desired capacity and static head with two or more fans of the same design but of different sizes and operating at different speeds. Frequently the fan that operates most efficiently under the given conditions is not the lowest in first cost, and the selection must be governed by the relative magnitude of these factors. But the controlling factor in fan selection often is, or should be, quietness of operation, which means a relatively low tip speed and a larger fan than the minimum possible size. In general, a fan will be quiet when it is operating at a speed not greater than that corresponding to its best efficiency. The subject of noise will be dealt with in more detail later.

Table 71 gives the outlet velocities and tip speeds that are good practice from the standpoint of efficient operation. Lower speeds, however, may be required where particularly quiet operation is needed.

TABLE 71.—OUTLET VELOCITIES AND TIP SPEEDS FOR EFFICIENT OPERATION

Static pressure, in. of water	Forward-curved blades		Backward- or double-curved blades	
	Outlet velocity, ft. per min.	Tip speed, ft. per min.	Outlet velocity, ft. per min.	Tip speed, ft. per min.
½	1,100	2,000	1,100	3,700
1	1,600	2,450	1,600	5,200
1½	1,900	3,650	1,800	6,350
2	2,200	4,250	2,200	7,850
2½	2,400	4,750	2,400	8,250

277. Fan Tables.—The exact performance to be expected of a fan under any given conditions is given in tables published by the manufacturers. These tables give the speed and horsepower for various static pressures and capacities, at 70° and 29.92 in. barometric pressure.

Table 72 is a portion of the table published for one size of a commercial type of fan.

Before selecting a fan the designer of a ventilating system will naturally have computed the volume of air to be handled, its temperature, and the static head against which the fan must operate. The proper size of fan must be selected to give a suitable outlet velocity, and it will often be desirable to select two or more fans of different sizes and compare the costs and the horsepower consumption to arrive at the best selection.

The choice of the outlet velocity is important chiefly from the standpoint of noise. In ventilating work it is well to limit the outlet velocity for office and public buildings to 1,400 and in extreme cases 1,200 ft. per min., where absolute quietness is essential or where the fan is connected to the room by only a short discharge duct.

Example 1.—Given: Static pressure 1¼ in., volume 14,500 c.f.m., temperature 70°. What will be the fan size, speed, and horsepower?

It will be seen from Table 72 that a No. 6 fan will handle 14,930 c.f.m. at a speed of 556 r.p.m. and an outlet velocity of 1,200 ft. per min., with a power consumption of 3.99 hp.

Example 2.—Given: Static pressure 1¼ in., volume 14,500 c.f.m., temperature 120°. Find the speed and horsepower from Table 72.

Sometimes air quantities are stated as the actual volume at the stated temperature, sometimes as the equivalent volume (equal weight) at 70°.

One should always make certain which is meant. In this case 14,500 is the actual volume at 120°.

Use rule 8 (Art. 275), which says that for a constant weight of air the capacity, speed, and pressure vary inversely as the density. The equivalent volume at 70° for the same weight of air is

$$14{,}500 \times \frac{459.7 + 70}{459.7 + 120} = 13{,}300 \text{ c.f.m.}$$

The corresponding pressure is $1.25 \times \frac{459.7 + 70}{459.7 + 120} = 1.145$ in. From Table 72 and by interpolation it will be found that a No. 6 fan will handle this 13,300 c.f.m. at 1.145 in. at a speed of 527 r.p.m. and a horsepower consumption of 3.26. At 120° the speed would be

$$527 \times \frac{459.7 + 120}{459.7 + 70} = 575 \text{ r.p.m.}$$

The horsepower would be (rule 9, Art. 275)

$$3.26 \times \left(\frac{459.7 + 120}{459.7 + 70}\right)^2 = 3.90 \text{ hp.}$$

Thus the No. 6 fan would deliver the 14,500 c.f.m. at 120° at a pressure of 1¼ in. while running at 575 r.p.m. and drawing 3.90 hp.

278. Fan and System Characteristics.—The static head developed by the fan for a given air delivery must obviously be the same as the static resistance of the duct system for the same flow. This relationship is illustrated in Fig. 225. The frictional resistance of a duct system varies nearly as the square of the flow, and the characteristic curves for the system are therefore parabolic. For a given speed, curve 1 represents the fan characteristic; and for a given setting of dampers, etc., curve 2 represents the system characteristic. Their intersection *A* therefore indicates the volume that will be delivered.

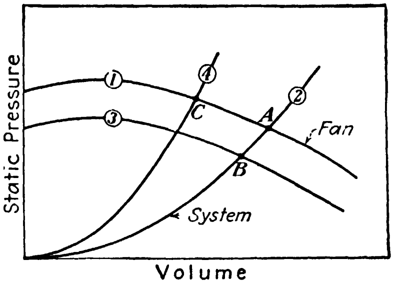

FIG. 225.—Fan and system characteristics.

If the fan speed is reduced to that corresponding to curve 3, then the abscissa of the point *B* represents the new volume. On the other hand, if without altering the fan speed the resistance of the system is increased by damping so that the system char-

TABLE 72.—TYPICAL FAN TABLE[1]
(American High Speed No. 6 Fan)

Double inlet Double width

Backward-inclined-blade type fan with nonoverloading power characteristic

Outlet 12.44 sq. ft. Wheel 3.000 ft. diameter Maximum brake hp. = $25.51 \left(\frac{\text{r.p.m.}}{1{,}000}\right)^3$

Volume, c.f.m.	Outlet velocity, f.p.m	⅞-in. r.p.[2]			1-in. r.p.			1¼-in. r.p.			1½-in. r.p.			1¾-in. r.p.			2-in. r.p.		
		Tip speed, f.p.m.	R. p.m.	B. hp.	Tip speed, f.p.m.	R. p.m.	B. hp.	Tip speed, f.p.m.	R. p.m.	B. hp.	Tip speed. f.p.m.	R. p.m.	B. hp.	Tip speed, f.p.m.	R. p.m.	B. hp.	Tip speed, f.p.m.	R. p.m.	B. hp.
7,464	600	4,216	447	1.45	4,480	475	1.79	4,992	530	2.35	5,460	579	2.98						
8,708	700	4,240	450	1.67	4,512	479	1.99	5,016	532	2.56	5,480	581	3.25	5,905	627	3.89	6,308	669	4.50
9,952	800	4,276	454	1.90	4,544	482	2.19	5,048	536	2.82	5,508	584	3.44	5,928	629	4.17	6,328	671	4.95
11,196	900	4,338	459	2.11	4,592	487	2.42	5,084	539	3.08	5,544	588	3.72	5,964	633	4.50	6,352	674	5.27
12,440	1,000	4,384	465	2.33	4,644	493	2.66	5,132	544	3.36	5,576	592	4.09	5,996	636	4.83	6,388	678	5.55
13,680	1,100	4,480	475	2.58	4,712	500	2.93	5,180	550	3.67	5,620	596	4.44	6,036	640	5.23	6,416	681	6.06
14,930	1,200	4,560	484	2.86	4,800	509	3.22	5,244	556	3.99	5,668	601	4.81	6,080	645	5.63	6,452	685	6.52
16,170	1,300	4,668	495	3.17	4,910	521	3.55	5,320	564	4.35	5,740	609	5.19	6,128	650	6.07	6,512	691	6.96
17,420	1,400	4,840	514	3.52	5,068	538	3.91	5,424	576	4.74	5,820	618	5.60	6,188	657	6.52	6,556	696	7.35
18,660	1,500	5,044	535	3.89	5,180	549	4.32	5,540	588	5.17	5,896	626	6.06	6,264	665	7.01	6,624	703	7.99
19,900	1,600	5,220	554	4.31	5,384	571	4.74	5,712	606	5.63	6,024	639	6.56	6,340	673	7.53	6,692	710	8.54
21,150	1,700	5,396	573	4.76	5,576	592	5.22	5,892	626	6.16	6,176	655	7.11	6,472	686	8.10	6,784	720	9.13
22,390	1,800	5,564	590	5.24	5,736	608	5.71	6,088	646	6.72	6,340	673	7.69	6,640	705	8.75	6,888	731	9.80
23,640	1,900	5.776	613	5.80	5,904	626	6.27	6,236	661	7.29	6,508	690	8.33	6,760	717	9.36	7,032	746	10.48
24,880	2,000	5,960	632	6.35	6,132	651	6.89	6,412	680	7.96	6,684	709	9.01	6,940	736	10.10	7,180	762	11.22
26,120	2,100	6,176	655	7.00	6,308	669	7.54	6,604	701	8.63	6,856	727	9.74	7,100	753	10.84	7,364	781	12.03
27,370	2,200	6,388	678	7.68	6,532	693	8.23	6,752	716	9.34	7,016	744	10.49	7,276	772	11.65	7,524	798	12.88
28,610	2,300	6,592	699	8.40	6,736	714	9.00	6,988	741	10.15	7,180	762	11.26	7,460	792	12.52	7,672	815	13.81
29,860	2,400	6,800	721	9.18	6,956	738	9.77	7,180	762	10.98	7,390	784	12.20	7,628	809	13.43	7,864	834	14.72
32,340	2,600	7,216	766	10.91	7,360	781	11.55	7,596	806	12.85	7,788	826	14.08	7,984	847	15.42	8,208	871	16.79
34,830	2,800	.....	...	.....	7,796	827	13.68	8,028	852	14.98	8,192	869	16.37	8,404	892	17.97	8,604	913	18.90
37,320	3,000	.....	...	.....	.....	...	.....	8,412	893	17.25	8,612	914	18.77	8,816	935	20.51	8,992	954	21.68
39,810	3,200	.....	...	.....	.....	...	.....	.....	...	.....	9,060	961	21.55	9,248	981	23.03	9,432	1,001	24.68
42,300	3,400	.....	...	.....	.....	...	.....	.....	...	.....	9,452	1,003	24.53	9,684	1,027	26.07	9,816	1,041	27.74

Volume, c.f.m.	Outlet velocity, f.p.m.	2¼-in. r.p.			2½-in. r.p.			3-in. r.p.			3½-in. r.p.			4-in. r.p.			5-in. r.p.		
		Tip speed, f.p.m.	R. p.m.	B. hp.	Tip speed, f.p.m.	R. p.m.	B. hp.	Tip speed, f.p.m.	R. p.m.	B. hp.	Tip speed, f.p.m.	R. p.m.	B. hp.	Tip speed, f.p.m.	R. p.m.	B. hp.	Tip speed, f.p.m.	R. p.m.	B. hp.
8,708	700	6,668	707	5.92															
9,952	800	6,696	710	5.98	7,056	749	6.26	7,712	818	8.28									
11,196	900	6,720	713	6.02	7,080	751	6.89	7,740	821	8.46	8,332	884	10.73						
12,440	1,000	6,748	716	6.46	7,104	754	7.37	7,752	822	8.92	8,356	887	10.81	8,920	946	12.83			
13,680	1,100	6,780	719	6.92	7,132	757	7.81	7,780	825	9.60	8,376	889	11.44	8,944	949	13.46	9,668	1,058	18.00
14,930	1,200	6,820	724	7.40	7,168	760	8.33	7,812	829	10.17	8,412	893	12.04	8,968	951	14.30	9,996	1,061	18.39
16,170	1,300	6,856	727	7.93	7,200	764	8.86	7,824	833	10.82	8,440	896	12.85	8,992	954	14.94	10,032	1,064	18.56
17,420	1,400	6,908	733	8.43	7,240	768	9.54	7,876	837	11.53	8,472	899	13.65	9,024	957	15.80	10,060	1,067	19.55
18,660	1,500	6,968	739	9.00	7,296	774	10.07	7,924	841	12.16	8,504	902	14.40	9,060	961	16.65	10,080	1,070	20.79
19,990	1,600	7.032	746	9.58	7,364	781	10.62	7,976	846	12.85	8,544	906	15.18	9,092	965	17.56	10,116	1,073	22.14
21,150	1,700	7,104	754	10.21	7,416	787	11.32	8,024	851	13.60	8,596	912	16.03	9,128	968	18.50	10,136	1,075	23.48
22,390	1,800	7,204	764	10.89	7,512	797	12.02	8,092	858	14.42	8,668	920	16.83	9,192	975	19.38	10,168	1,079	24.83
23,640	1,900	7,292	774	11.59	7,576	804	12.73	8,148	864	15.18	8,712	924	17.65	9,228	979	20.30	10,208	1,083	25.84
24,880	2,000	7,428	788	12.37	7,700	817	13.56	8,232	873	16.04	8,768	930	18.63	9,288	985	21.33			
26,120	2,100	7,580	804	13.21	7,824	830	14.40	8,356	887	16.96	8,844	938	19.64	9,352	992	22.36			
27,370	2,200	7,732	820	14.07	7,964	845	15.35	8,432	895	17.91	8,920	946	20.62	9,424	1,000	23.45			
28,610	2,300	7,924	841	15.01	8,144	864	16.34	8,560	908	18.93	9,012	956	21.69	9,504	1,008	24.55			
29,860	2,400	8,064	856	15.98	8,296	880	17.29	8,732	926	20.08	9,148	971	22.89	9,595	1,018	25.76			
32,340	2,600	8,440	896	18.16	8,684	921	19.62	9,080	963	22.44	9,436	1,001	25.36	9,828	1,043	28.38			
34,830	2,800	8,776	931	20.54	9,008	956	22.00	9,392	996	25.03	9,804	1,040	28.14						
37,320	3,000	9.168	973	23.19	9,356	992	24.73	9,716	1,031	27.89									
39,810	3,200	9,572	1,016	26.12	9,716	1,031	27.77	10,064	1,068	31.02									
42,300	3,400	9,995	1,060	29.45															

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[1] Courtesy of the American Blower Corporation. Capacity tables are based on tests made in accordance with National Association of Fan Manufacturers' Code.

[2] R.p. is resistance pressure of system to flow of air in terms of inches water gage or static pressure on a blowing fan.

acteristic is represented by curve 4, then the point *C* represents the new pressure and volume.

When a reduction of delivery is desired, it can be accomplished with less loss of power by reducing the fan speed than by damping, but the latter method is usually more convenient unless the fan is equipped with some sort of variable-speed drive.

279. Axial-flow Fans.—In any axial-flow or propeller-type fan the air leaves the blades parallel to the wheel axis, as opposed to radial flow in a centrifugal fan. Actually, an axial-flow fan working against pressure gives the air leaving the blades a spiral or rotating motion. Stationary guide vanes, fitted either on the inlet or discharge of the fan, spiral the air in a direction opposite to the wheel rotation, and as a result the air leaves the fan in a true axial direction.

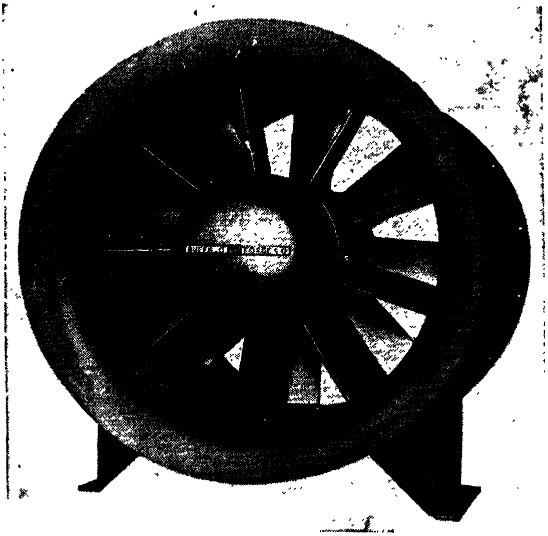

FIG. 226.—Axial flow fan. (*Buffalo Forge Co.*)

One characteristic of the axial-flow fan shown in Fig. 226 is the large hub and widely spaced airfoil blades. Actually that portion of the blade near the wheel center is practically ineffective, and so it is eliminated and replaced by the hub. Some wheel designs have hubs of 50 to 70 per cent of the wheel diameter. These large hubs permit locating the motor of direct-driven units, or the bearings in a belt-driven design, inside the fan without obstructing air flow.

Axial-flow fans show a similarity of shape in the static-pressure characteristic curve to that of the centrifugal fan with forward-curved blades. The axial-flow horsepower curve, however, is quite flat, showing a small rise near wide-open volume.

The principal advantage of an axial-flow fan is its inherent ability to handle large air volumes through a wide range of pressures in a small space with a straight-through uniform air flow, thus eliminating difficult duct connections.

280. Disk Fan.—The disk fan as illustrated in Fig. 227 is somewhat similar to the axial-flow fan, except that its design is particularly adapted for handling considerable quantities of air against very low pressures. It is therefore widely used where the air is moved into or from a room without passing through a system of ducts. When not used against too great a resistance (not more than ½ in. static), the disk fan is nearly as efficient as the centrifugal fan. The reason for the reduced efficiency against high pressures is that the axial pressure created by the portions of the blades near the center of the fan is relatively low, and there is a tendency for the air to flow radially inward rather than against the duct resistance.

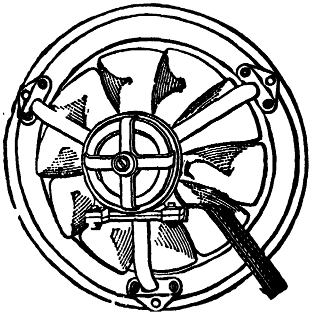

FIG. 227.—Disk fan.

The disk fan is governed by the same laws of performance (Art. 275) as the centrifugal fan. It fills a useful purpose in creating air movement in a room and is also much used for exhausting directly to the out-of-doors and for blowing air through certain designs of unit heaters installed without ductwork. The horsepower and pressure characteristics of disk fans are considerably different from those of centrifugal fans, as can be seen by comparing Fig. 228 with Fig. 224.

One serious objection to this type of fan is its noisiness. If this difficulty were eliminated, the disk fan would undoubtedly be much more widely used for low-head applications. Recently designed fans are less noisy than the older types. The requirements for quietness[1] include

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[1] See K. D. McMahan, "The Design of Quiet Propeller Fans," *Gen. Elec. Rev.*, February, 1934.

1. Increase of the usual low velocity at the fan center.
2. Elimination of contraction of the air stream by the fan.
3. A large effective width of blades.
4. Blade design to give uniform acceleration over width of blade and to prevent air from leaving or entering at periphery.

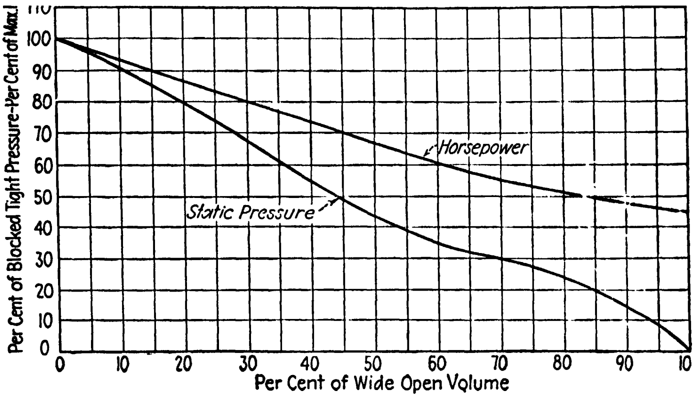

FIG. 228.—Characteristics of a disk fan.

281. Fan Drive.—In modern buildings most ventilating fans are driven by motors, but there are certain cases where steam-engine drive is more economical, as in schools and office buildings where high-pressure steam is available and where the engine exhaust can be used in the heating system. Under such conditions the saving in the cost of power may be sufficiently attractive to outweigh the objections of the greater amount of attention and maintenance required by the steam engine.

The engines built for fan drive are naturally not designed for particularly good steam economy, since the steam requirements of the heating system will ordinarily be far in excess of the exhaust steam available. The water rates of average simple slide-valve engines at rated loads are given in Table 73 on the basis of indicated horsepower. The brake horsepower may be determined by multiplying the indicated horsepower by the mechanical efficiency, which will vary from 90 per cent for very small engines to 95 per cent for the larger sizes. At loads below the rated load, the water rates naturally increase rather rapidly.

Steam turbines are occasionally used for fan drive, but because of their high speed they are not very suitable. In ventilating

TABLE 73.—WATER RATES OF SLIDE-VALVE ENGINES, STEAM PRESSURE 100 LB. GAGE

Rated Indicated Hp.	Water Rate at Rated Load (⅜ Cutoff), Lb. per I.Hp. per Hour
10	60
20	50
30	44
40	39
50	35
60	33
70	31
80	30

work, some form of reduction gear is usually required if turbine drive is used.

Steam drive is seldom feasible for fans that are to be operated in summer. Actually it is now used in only a few installations.

Motors for fan drive should be selected with a horsepower rating from 10 to 25 per cent in excess of the power requirement as taken from the fan tables. This is necessary because of the possibility of errors in computing the static pressure that might result if the fan has to work against a higher pressure than estimated.

In some kinds of fan systems it is desired to vary the flow of air from time to time; and as this can be done more efficiently by varying the fan speed than by using dampers, it is desirable to use variable-speed motors. The question of speed variation is simple if direct current is available, because d-c motors are available that operate at any ordinary fan speed and with a wide range of speed variation. Direct-current motors are often connected directly to the fan rather than through belting, which saves space and is a better arrangement. Where the fan speed is very low, however, it will usually be cheaper to use a higher speed motor and a belt drive.

Alternating-current motors are obtainable in certain standard speeds. In ventilating work they are usually connected to the fans by belts. For small fans single-phase motors are used, but three-phase motors are more desirable for sizes above about 5 hp. If a constant-speed motor is to be used, the squirrel-cage induction motor is suitable. For variable speed use the wound-rotor induction motor, which is capable of a speed reduction to as low as 50 per cent of synchronous speed. Standard speeds for alternating-current motors are 1,750, 1,150, 865, 690, and 570 r.p.m. Motor efficiencies for d-c motors are approximately as follows: 85 per cent for 5 hp., 89 per cent for 25 hp., and 91 per cent for 100 hp. The efficiencies of a-c motors are 1 to 3 per cent lower than the foregoing values.

Slow-speed a-c motors are considerably more expensive than high-speed motors, are noisier, and have a poorer power factor, so that it is usually better to use a higher speed motor with a suitable pulley ratio, which should not be greater than 6:1.

Speed control may also be accomplished by the use of a variable-speed hydraulic coupling, which allows the motor to operate at constant rotation. At part load most motors maintain good efficiency, but under such conditions coupling efficiency decreases, so that the over-all efficiency of direct-current operation is rarely attained.

Another means of operating a fan at constant speed is to use variable guide vanes on the inlet to the fan. Air horsepower is reduced by causing the air to enter the wheel with a tangential component in the direction of wheel rotation, which reduces the net static pressure developed by the fan. This method of fan-inlet regulation is better than damping at the fan outlet for air-volume control.

AIR-CLEANING DEVICES

282. Types.—There are three methods in use for removing dust from air in air-conditioning work—the filter, the electric precipitator, and the spray washer. The washer is not so efficient as the other devices and is not often installed as a cleaning device primarily. The other two devices will be considered first.

Filters are of two general types—those that actually screen the dirt out of the air by passing it through materials such as cotton or cellulose fiber, and those in which the air impinges upon surfaces coated with viscous oils.

Figure 229 shows the former type, in which the fabric is arranged zigzag fashion in metal frames. The fabric is renewed when it becomes so loaded with dirt that the air resistance builds up to an undesirable point.

This type of filter is quite efficient. It is used principally in the larger installations, where it can be properly maintained. The face velocities recommended are 350 to 500 ft. per min., at which the pressure loss through a clean filter is from 0.10 to 1.15

in. The fabric is usually renewed when the pressure loss has increased to 0.5 in. or thereabouts.

283. The Viscous Filter.—The viscous filter consists of a porous layer of material coated with a viscous substance, usually oil. The materials used are metal ferrules or shavings, glass fibers, and the like. The object is to cause the air to make several changes of direction as it flows through, so that the dirt will impinge on the surfaces and will be entrapped by the viscous fluid.

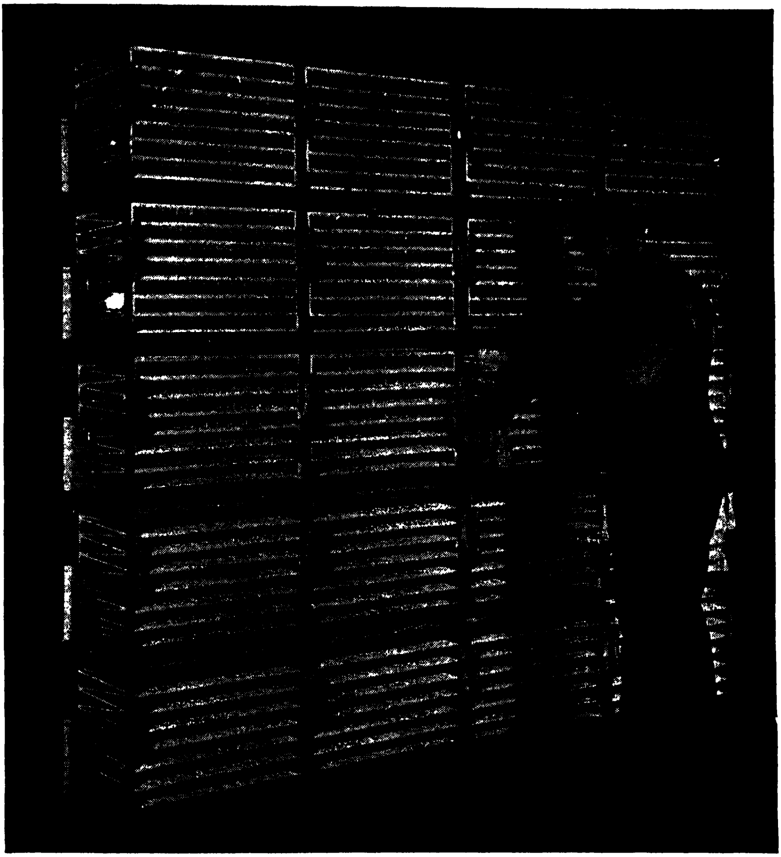

FIG. 229.—Dry-fabric type of filter. (*Independent Air Filter Company.*)

One form is shown in Fig. 230. Between two panels of expanded metal screen, enclosed in a frame of convenient size for handling, are metal shavings or stampings (almost any shape will do). The units are dipped in oil, which coats the metal and traps the impinging dust as the air flows through the filter. A complete filter consists of as many of these units as are necessary to give the requisite face area.

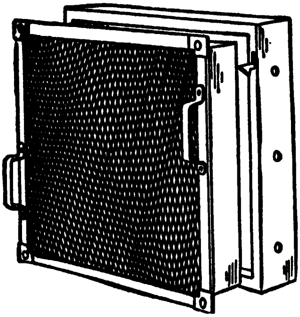

FIG. 230.—Viscous filter unit.

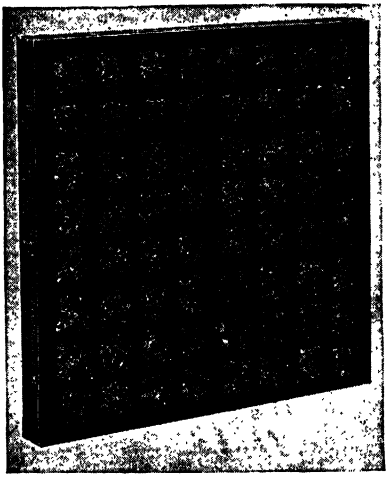

FIG. 231.—Throwaway type of filter. (*American Air Filter Company, Inc.*)

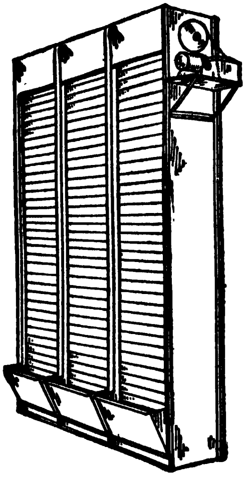

FIG. 232.—Belt-type viscous filter.

Several recent designs of filter units are supposed to be thrown away when they become loaded with dirt. Figure 231 shows a unit of this type. The casing is of heavy cardboard or fiber. Most of the unit air conditioners are equipped with throwaway filters. These have superseded to a large extent the type shown in Fig. 230, which involves a disagreeable job of cleaning.

The dimensions of the throwaway types of filter have been largely standardized to fit into a 20- by 20-in. frame. The usual recommended capacity is 800 c.f.m. per unit, equivalent to a face velocity of 280 ft. per min. Where several units are required, they are arranged zigzag fashion to conserve space.

The belt type of filter (Fig. 232) has also been developed to eliminate the more or less troublesome operation of periodically cleaning filter units like those shown in Fig. 230. An endless belt made up of filter sections is supported from a horizontal shaft that is periodically revolved by a small motor, so that the filter sections are immersed in the tank of oil at the bottom of the apparatus and are thereby cleaned and recoated with oil. The operation may be made automatic by means of a time clock in the motor circuit. One objection to this type of filter is the presence of a considerable quantity of combustible oil near the air stream.

Since filters of this type operate on the impingement principle, it is evident that much of the dirt will be entrapped near the upstream face and that progressively decreasing amounts will be entrapped toward the downstream side. In fact, the *percentage* of the total solids remaining in the air that is removed by each successive impingement remains approximately the same. A properly designed filter has larger voids near the entering face to take care of the accumulation of dirt and a decreasing amount of space toward the downstream face.

284. Rating and Testing.—The A.S.H. & V.E. Code[1] gives a method for testing filters and rating them. A standard dust of specified composition is introduced into the air stream while the filter is operating at rated capacity. The **dust arrestance,** usually expressed in per cent,[2] is determined by the formula

$$E = 1 - \frac{G_1}{G_0} \tag{12}$$

where E = arrestance.

G_1 = dust concentration beyond the filter.

G_0 = dust concentration ahead of the filter.

The dust-holding capacity of a nonautomatic filter is the amount by weight of standard dust that the filter will hold without exceeding the following resistances:

0.18 in. for low-resistance type
0.50 in. for medium-resistance type
1.00 in. for high-resistance type

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[1] Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work, *Trans. A.S.H. & V.E.*, 1933.

[2] *I.e.*, by multiplying by 100 (not so stated in Code).

These resistance limits serve to classify filters. The low-resistance type is usually used in warm-air furnace work and in unit air conditioners. The medium-resistance type is usually specified for central fan systems. The high-resistance type is more suitable for such services as the intakes of air compressors than for general air-conditioning work.

The National Bureau of Standards[1] has devised a *blackness* test for establishing the efficiency of an air-cleaning device. This test disregards particle weight and instead establishes the degree of discoloration or blackness with air samples drawn through filter papers before and after cleaning as the criterion of cleaning efficiency. A photometer is used to compare the degree of blackness on the filter papers.

When it is realized that dust-particle sizes in ventilating and air-conditioning systems may range from 0.1 to 100 microns,[2] the difficulty of devising a performance standard to cover all conditions of service becomes evident. A tobacco-smoke particle is about 0.1 micron in size, a coal-dust particle 10 microns, and a ragweed pollen grain 20 microns, whereas a particle must be 50 microns or more in diameter to be visible to the naked eye.

285. Electrostatic Cleaners.—The inability of a dry or viscous type of air filter to remove fine dust particles and tobacco smoke has led to the development of the electrical precipitator, which is particularly efficient in removing the smaller particles. The precipitation method consists of imparting an electrical charge to each dust particle by passing the air between electrodes and then collecting the dust on parallel plates as the air flows between them. Figure 233 shows the diagrammatic arrangement. A d-c voltage of 12,000 to 15,000 is impressed on the wires *A*, *A*. The cylinders *B*, *B* are grounded. This potential difference causes a flow of ions between the wires and the cylinders, and particles in the air stream become charged.

The plates shown by dotted lines are charged with about one-half of the voltage on the wires *A*, *A*. The electrostatic field created between the charged and grounded plates drives the dust particles to the latter, where they accumulate. The plates are frequently coated with a thin film of oil to ensure that the dust particles will adhere. Cleaning is accomplished by periodically flushing with water. Electronic tubes are used to step up the voltage from that of the ordinary power supply.

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[1] R. S. Dill, "A Test Method for Air Filters," *Trans. A.S.H. & V.E.*, 1938.

[2] One micron equals 1/25,000 in., or the equivalent of about 1/50 of the diameter of the human hair.

An installation requires multiples of a unit collector cell in proper number to handle the air-volume requirements. These cells are made in two standard sizes, 18 and 36 in. long or high by 8½ in. wide and 31 in. deep in the direction of air flow. At the maximum rated velocity of 375 ft. per min. the resistance is about .05 in. of water.

This type of cleaner can be constructed to fit into the usual air-conditioning system. The precipitator does not entirely displace other air filters at the present time because of its relatively high cost.

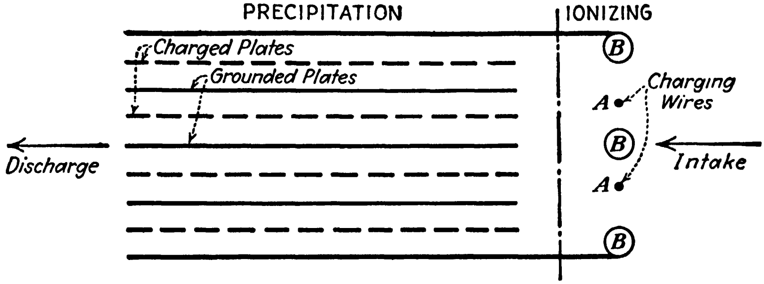

FIG. 233.—Diagram of electrostatic precipitator. (*Westinghouse Manufacturing Corporation.*)

This device, along with other air-purification equipment discussed in Art. 286, is useful in reducing the amount of outdoor air required for ventilation purposes. In restaurants it is particularly suitable for removing smoke where it is desired to recirculate air for the purpose of conserving fuel and also to reduce the heating- or cooling-load requirements.

286. Air Purification.—As mentioned in Chap. XIV, many theories have been advanced for purifying the air. However, except for apparatus for removing dust particles and odors, very little practical equipment has been designed for producing air free of bacteria.

It is claimed that most air-borne bacteria are conveyed by dust particles, and for that reason electrostatic precipitation, having a high efficiency in removing minute dust particles, is also thought to be efficient in the removal of bacteria.

Ultraviolet lamps have been installed in a few hospitals, nurseries, and schools to sterilize the air. The ultimate success of such applications depends upon locating the lamps directly in the space to be treated and shielding the rays so that they will not strike the eyes or skin of the occupant.

Apparatus for producing germicidal mists[1] of *triethylene glycol* have also been suggested for use in connection with the customary ventilation system to reduce the bacterial contamination of a space. One method of introduction mixes the glycol with water, and the resulting solution is atomized by spray nozzles into a moving air stream. Atmospheres containing triethylene glycol are invisible, odorless, and nonirritating. Relative humidities in the range from 25 to 60 per cent are required for optimum bactericidal action.

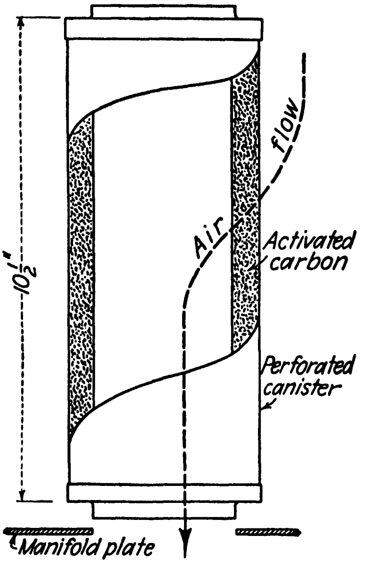

FIG. 234.—Diagram of activated carbon canister. (*W. B. Connor Engineering Corp.*)

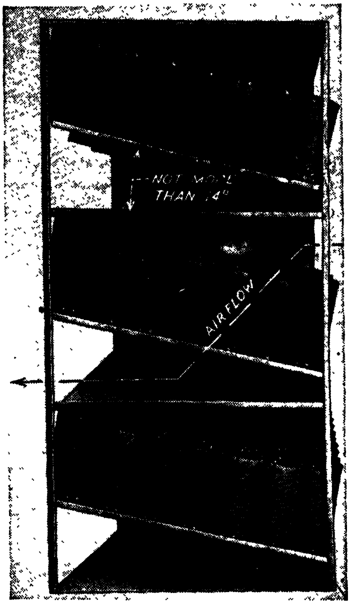

FIG. 235.—Typical assembly arrangement of activated carbon canisters. (*W. B. Connor Engineering Corp.*)

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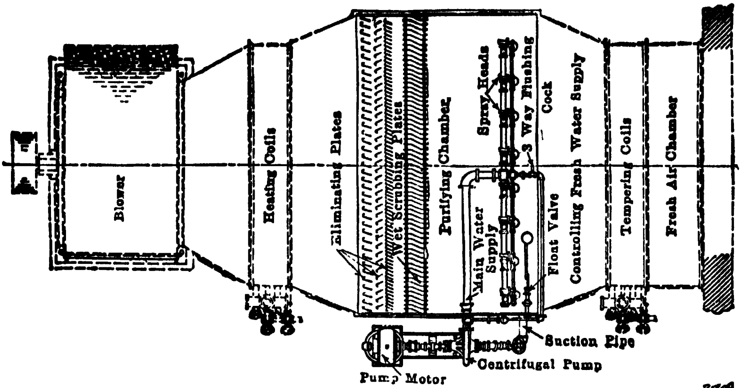
[1] JENNINGS, BIGG and OLSEN, "The Use of Glycol Vapors for Air Sterilization and the Control of Air Borne Injection," *Jour. A.S.H. & V.E.*, September, 1944.

Air-contaminating impurities are constantly being released within closed spaces in the form of objectionable odors from human body emanations, tobacco smoke, cosmetics, or food preparations. These odorous and objectionable gases and vapors may be removed by adsorbing them in *activated carbon.*

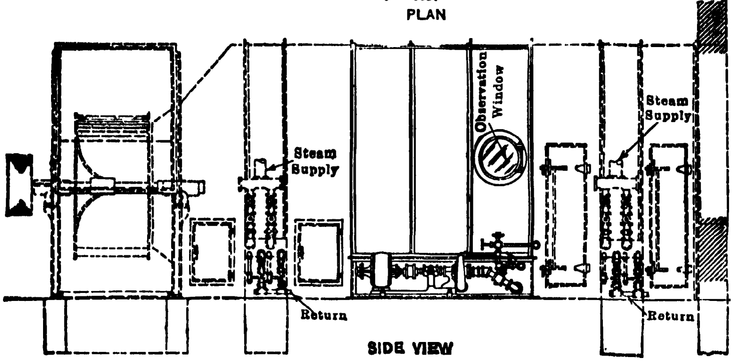
The air-purifying effectiveness of activated carbon depends upon the duration of contact between the air and carbon. It has been found that for residential applications approximately 25 lb. of carbon are required per 1,000 c.f.m. of air treated and that 35 lb. are needed for the average commercial installation. Typical installations consist of perforated cylindrical canisters filled with carbon, as illustrated in Fig. 234, which are assembled in multiple on manifold plates, as shown in Fig. 235. The recommended air capacity of each canister varies from 20 to 35 c.f.m., and the resistance to air flow in this range will be from 0.075 to 0.25 in. of water.

It is advisable to provide filters on the air-inlet side to protect the activated carbon equipment from dust accumulation, which would reduce its effectiveness. After 1 or 2 years of service, depending upon the application, the canisters must be removed for reactivation. A temperature of about 1000° is usually required for a complete release of all the gases and vapors.

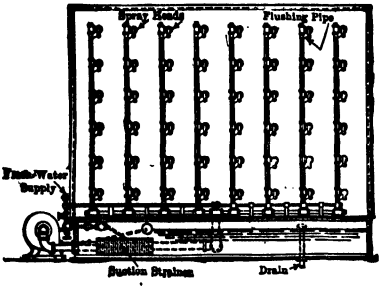
287. The Air Washer.—A typical air washer is shown in Fig. 236. It consists of three elements—the spray nozzles, the scrubber plates, and the eliminator plates. The nozzles are placed in a bank across the path of the air, and the water is forced through them by a pump and is discharged in a fine conical spray or mist in the direction of the air flow. In some cases two or three banks of nozzles are used. The air, drawn through the washer by the fan, is thus brought into intimate contact with the water, and some of the dirt and soluble gases are removed. The real cleansing, however, is done by the scrubber plates, which are designed to change the direction of flow so that the dirt will be thrown out from the air by its inertia and by the rubbing of the air over the wet surfaces. The plates are kept flooded either by the spray nozzles or by a separate row of nozzles placed above them. Following the scrubber plates is a series of eliminator plates, whose function is to remove the entrained water from the air. The lower part of the washer constitutes a tank into which the water falls and from which it is taken by the circulating pump.



PLAN



SIDE VIEW



END VIEW

FIG. 236.—Air washer.

A float valve admits fresh water as required to replace that evaporated.

Proper provision must be made in an air washer to prevent trouble from the large quantities of dirt that are washed from the air and deposited in the tank. A screen of ample area is necessary on the suction line to the pump to prevent the dirt from being carried into the circulating system, and in some types of washers special devices are necessary to enable the spray nozzles to be cleaned periodically by flushing. The accumulated dirt must be removed from the tank at frequent intervals.

In a ventilating system where the outside temperature falls below the freezing point it is quite necessary to protect the air washer from freezing by a tempering heater placed ahead of it.

The air washer is fairly effective in cleansing the air of dust and has two other very important functions. It can be used as a humidifier or as a dehumidifier and cooler and as such is valuable in air conditioning. Its use as an air cleaner is decreasing.

288. Humidification.—A moderate amount of humidification can be obtained in an air washer when unheated spray water is used, but to humidify sufficiently and with proper control in winter it is necessary to warm the spray water to a temperature depending upon the amount of humidity desired or else to raise the temperature of the entering air considerably. The usual method is to heat the incoming air to about 40° and to heat the spray water. If sufficient heat is applied, the air leaving the washer will be in a fully saturated condition and at a higher temperature than the entering air. The amount of humidity may thus be readily controlled as desired.

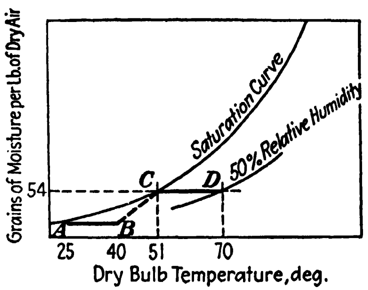


FIG. 237.

The humidifying action will be more clearly understood by referring to the diagram of Fig. 237, which is a reproduction of a portion of the psychrometric chart. Assume that the air is drawn from outside at a temperature of 25° and a relative humidity of 80 per cent, as represented by point *A*. In passing through the tempering heater it is heated to 40°, as represented by point *B*, and enters the air washer. Here both the temperature and the moisture content are further increased until the air reaches saturation at point *C*. Assume that the final condition of the air that is desired is 70° and 50 per cent relative humidity, requiring a moisture content of 54 gr. per lb. of dry air. It would then be necessary for the air to leave the washer in the condition represented by point *C*, with 54 gr. of moisture per lb. and at a temperature of 51°. Further heating after leaving the washer will bring it to the desired temperature of 70°, as represented by point *D*. The exact shape of the line *BC* is not known but is unimportant.

289. Humidity Control. The fact that the air always leaves a washer, in which the spray water is sufficiently heated, in a *saturated condition* makes possible a simple method of humidity control known as the **dew-point method.** The moisture content of the air at saturation depends upon its temperature, the relation being shown by the saturation curve of the psychrometric chart. For every moisture content there is a definite saturation or dew-point temperature. In the foregoing example it was 51°. Therefore, if the temperature of the air leaving the washer is regulated, the moisture content will be accurately controlled.

In practice this regulation of temperature is accomplished by a thermostat, which is placed in the path of the air leaving the washer and controls the amount of heat added to the washer water so that the air leaves the washer at the proper dew-point temperature.

Another method of humidity control involves the use of a humidistat or hygrostat located in one of the ventilated rooms. It is similar to an ordinary compressed-air thermostat except that the temperature element is replaced with a material such as cross-grained wood or hair, which undergoes a considerable change of dimension under a varying moisture content in the surrounding air. The humidistat, like the dew-point thermostat, controls the amount of heat added to the spray water.

When a considerable portion of the air passing through the washer is recirculated air, there is danger that the humidity will build up to an undesirable point. The way to avoid this is to mix fresh air and recirculated air in such proportions that the mixture will be at a temperature of 40 to 45°. If more recirculation is desired, shut down the spray nozzles in the washer entirely or arrange the system in such a way that the recirculated air does not pass through the washer.

290. Spray-water Heater.—The spray water is heated externally to the washer in a heater, the steam supply of which is controlled by the dew-point thermostat or humidistat. If high-pressure steam is available, an injector type of heater (Fig. 238) is used. Steam is mixed with the water to the proper amount in a mixing nozzle, the amount of steam being controlled by a valve operated by compressed air. The dew-point thermostat or humidistat controls the amount of valve opening.

When only low-pressure steam is available, or in a hot-water heating system where it is desired to use the circulating water to heat the spray water, a surface heater is used, and the steam or hot-water supply to it is controlled in a similar manner.

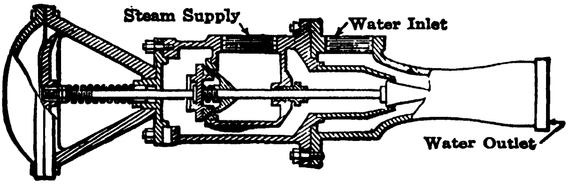


Fig. 238.—Spray-water heater.

291. Humidifying Efficiency with Unheated Water.—When the spray water is *not heated* by external means, it tends to attain the wet-bulb temperature of the air passage through the washer. The amount of humidification of the air is a function of the intimacy and time of contact between the water and the air. When the maximum amount of humidification is desired without heating the water, additional banks of nozzles are provided. The direction of the spray with respect to the air flow also has an influence.

The **humidifying efficiency** of a washer is the ratio, in per cent, of the dry-bulb reduction to the wet-bulb depression. The following humidifying efficiencies[1] are to be expected with various spray-bank arrangements:

	Saturation Efficiency, Per Cent
3 Banks: 2 upstream, 1 downstream	100
2 Banks: 2 upstream	95
2 Banks: 1 upstream, 1 downstream	85
1 Bank: upstream	80
1 Bank: downstream	65

[1] A.S.H. & V.E. "Guide."

In ordinary practice, for winter humidification the spray water is heated in preference to the method of providing additional banks of sprays to assure humidification.

292. Cooling by Humidification.—If no heat is added to the spray water of an air washer, some evaporation will still take place, but the latent heat of the vaporization is taken from the air itself, and the temperature of the air is consequently lowered. The extent of the cooling effect depends upon the capacity of the entering air for absorbing moisture or, in other words, upon the **wet-bulb depression** of the entering air. As the air absorbs moisture in the spray chamber, its dry-bulb temperature drops, but the wet-bulb temperature, which is a measure of the total heat of the mixture, remains unchanged. If the water is recirculated, its temperature soon drops to the wet-bulb temperature. The cooling effect actually obtained depends upon the humidifying efficiency of the washer. Referring to the psychrometric chart in Fig. 239, the point A represents the original condition of the air at 90° dry-bulb temperature and 75° wet-bulb temperature. The cooling and humidifying action is represented by the constant wet-bulb temperature line AB, the point B representing the final condition of 79.5° dry-bulb temperature. The humidifying efficiency of the washer is then $\frac{90 - 79.5}{90 - 75} = 70$ per cent, and the amount of moisture is increased from 105 to 123 gr. per lb. of dry air.

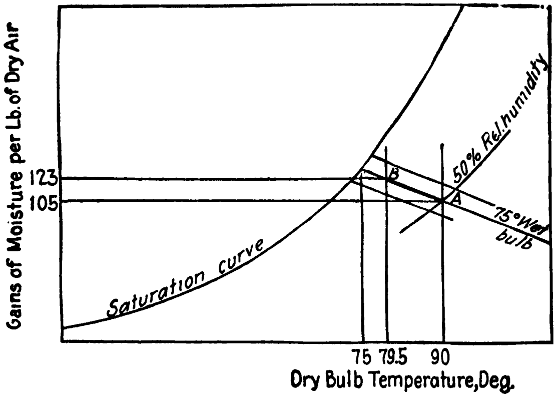


FIG. 239.

293. Other Forms of Humidifiers.—In fan heating systems it is quite feasible to humidify the air by the simple expedient of installing a steam jet in the air duct at the fan. This method is open to the objection, however, that an unpleasant odor often accompanies the steam.

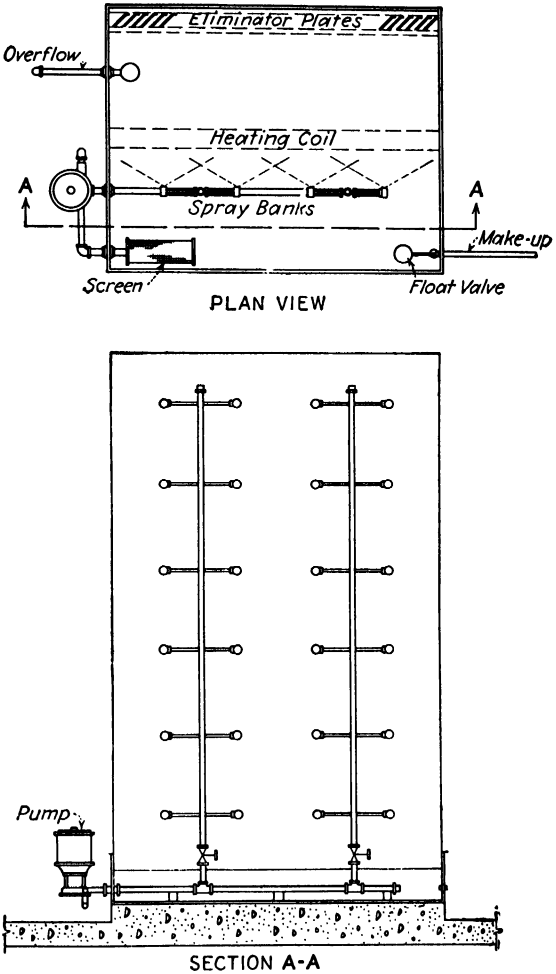


Fig. 240.—Humidification by means of water spray on the heating coil.

A better method is to spray water directly on the heating coils, provided they are of the nonferrous type so that they do not corrode. As shown in Fig. 240, this is accomplished by a simple bank of nozzles on the upstream side of the coil, with a small pump supplying them from the tank below, into which the surplus water drains. Automatic control can be applied by simply arranging a humidistat to start and stop the pump.

A simple **mechanical humidifier** used in industrial air conditioning is that shown in Fig. 241. Compressed air atomizes the water into a fine spray.

In the humidifier of Fig. 242 the air is drawn downward by a fan through a water spray, and the moistened air is projected horizontally in all directions. These mechanical humidifiers are used principally in industrial air conditioning.

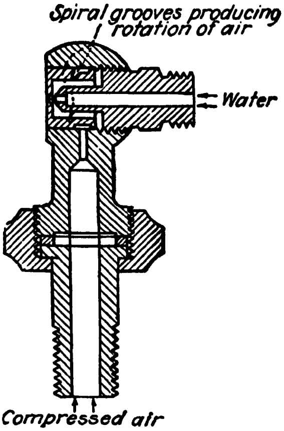


Fig. 241.—Atomizing humidifier.

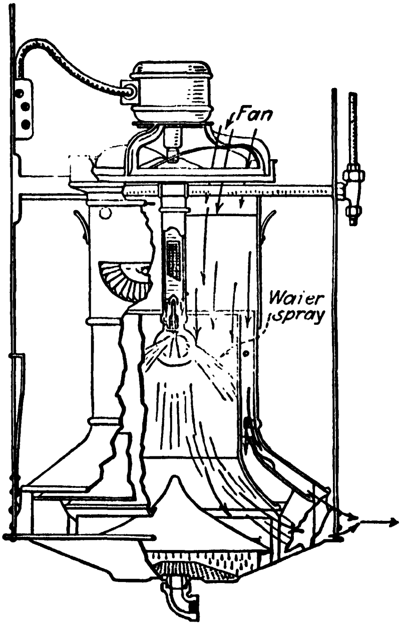


Fig. 242.—Mechanical humidifier with fan.

294. The Washer as a Dehumidifier and Cooler.—The air washer is a satisfactory apparatus for cooling and dehumidifying and is used in comfort cooling and for industrial air conditioning. For such purposes the washer is provided with not less than two banks of sprays. The spray water may be cooled externally, or a cooling coil containing brine or a refrigerant may be provided in the spray chamber of the washer.

When used as a dehumidifier and cooler, the washer extracts heat from the air by contact between the water and air. The air leaves the washer in a condition of saturation and with its wet-bulb temperature lowered by an amount depending upon the washer design and the water temperature.

295. Summary.—The designer of an air-conditioning system has the choice of using a filter, an electrostatic precipitator, or an air washer for the cleaning operation. Their relative merits for various applications may now be examined.

The advantages of the filter are that it is noiseless in operation, less subject to corrosion, simpler to maintain, requires less space, and is often less costly than the washer. Also, it will remove some kinds of dirt that will pass through a washer.

The advantages of the washer are that it will accomplish a fair job of cleaning and can also be used for humidifying and for cooling and dehumidifying. With a filter the cooling and dehumidifying are often accomplished by coils, and humidifying must be done by an additional piece of equipment.

In some cases both a filter and an air washer are installed to provide maximum cleaning plus humidification.

The precipitator is more costly but has a higher cleaning efficiency, especially for very small particles. It has not been available long enough to judge its popularity.

In general the present trend is toward filters in all but the larger installations.

HEATING AND COOLING COILS

296. Heaters.—Three types of heaters are used in fan systems: the cast-iron heater, the wrought-iron pipe coil, and the non-ferrous fin-tube element. The pipe coil was once in common use but is now obsolescent. The cast-iron heater is made up of sections, as shown in Fig. 243, connected at the top and bottom by right- and left-hand nipples cast with a hexagonal nut at the middle. A row of sections thus connected constitutes a **stack.** The sections are obtainable in nominal lengths of 30, 40, 50, 60, and 72 in. All sizes are connected at both top and bottom and are therefore suitable for hot water as well as steam. The sections are furnished in two heights, the regular and the narrow, and by the use of nipples of different lengths the distance between sections can be made 4⅝, 5, or 5⅜ in. center to center, the 5-in. spacing being standard. The surfaces are broken up by a large number of projections that extend into the air passages and serve to augment the heating surface in an effective manner. The principal dimensions of the sections of various sizes are given in Table 74.

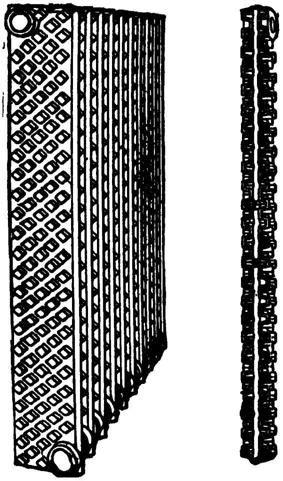


FIG. 243.—Cast-iron heater.

The method of installing the stacks in a sheet-metal casing is shown in Fig. 244. The stacks are staggered so as to break up the streamlines and increase the intimacy of the contact between the air and the heating surface. The spaces left at the ends of the stacks by the staggered arrangement are partially closed by strips of angle iron.

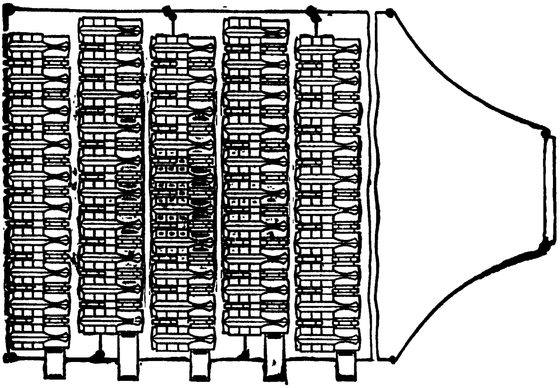


FIG. 244.—Vento heater installed in casing.

TABLE 74.—DIMENSIONS OF VENTO SECTIONS, INCHES

Nominal size		Square feet of surface	Actual height	Width
Regular	30	8.00	30	$9\frac{1}{8}$
	40	10.75	$41\frac{1}{64}$	$9\frac{1}{8}$
	50	13.50	$50\frac{9}{32}$	$9\frac{1}{8}$
	60	16.00	$60\frac{11}{16}$	$9\frac{1}{8}$
	72	19.00	$72\frac{3}{32}$	$9\frac{1}{8}$
Narrow	40	7.50	$41\frac{1}{64}$	$6\frac{3}{4}$
	50	9.50	$50\frac{29}{32}$	$6\frac{3}{4}$
	60	11.00	$60\frac{11}{16}$	$6\frac{3}{4}$

Approximate weight 8.2 lb. per square foot of surface.

297. Fin-tube Heating Units.—The use of nonferrous fin-tube heating and cooling units has become widespread, and they are largely superseding the cast-iron type. There are several forms of fin tubes, one of which is shown in Fig. 245. This particular style is made by wrapping a strip of copper edgewise around a copper tube, the inner edge being crimped in the process. Solder is then applied to fasten the fin in place and to ensure good heat conduction from the tube to the fin. The units are assembled by rolling the tubes into tube plates that form part of the supply headers. Figure 246 shows a complete unit. The sides of the several units when bolted together form the only housing that is required.

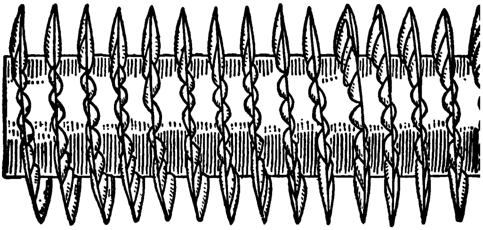


FIG. 245.—Section of fin tube.

The advantages of the nonferrous unit are its lightness (about one-tenth as heavy as cast-iron units) and the fact that less space is required than with the other forms. It is also built for high-pressure steam.

298. Heat Transmission.—In a fan system the depth of the heating unit depends upon the temperature rise through which the air is to be heated. Since the rate of heat transmission varies nearly as the temperature difference between the steam and the air, the heat transmitted from the last units is much less than from those with which the cold air first comes into contact.

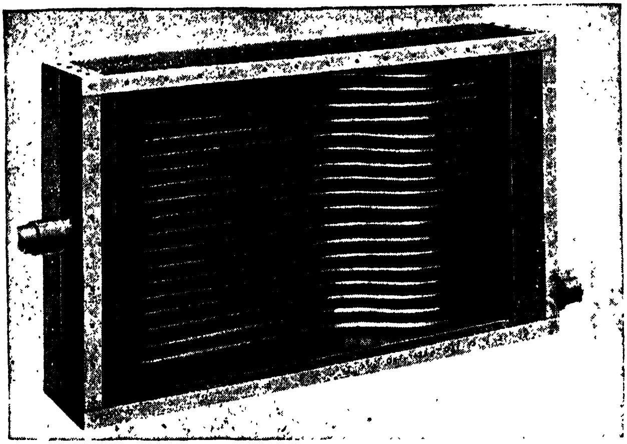


FIG. 246.—Nonferrous, fin-tube unit. (*Aerofin Corporation.*)

The final temperature to which the air is heated depends upon the number of units through which the air passes in series and upon the velocity of the air. The cross-sectional area of the heater depends upon the quantity of air delivered, the stacks or units being chosen of sufficient size so that the **free area** between the sections will allow that quantity to pass through at the velocity chosen. The free area per section for Vento heaters is given in Table 75. The velocity through nonferrous units is computed on the basis of face area.

TABLE 75.—FREE AREAS OF VENTO SECTIONS

Size of section, inches	Free area, square feet per section		
	5⅜-in. centers	5-inch centers	4⅝-inch centers
30	0.542	0.460	0.390
40	0.720	0.620	0.525
50	0.905	0.768	0.650
60	1.085	0.921	0.781
72	1.303	1.104	0.937

The velocity to be assumed depends upon the nature of the installation. In public buildings and in other places where the noise that accompanies high velocities is objectionable, the velocity through the free area of the heater should be limited to between 1,000 and 1,300 ft. per min., whereas in factories and similar buildings a velocity between 1,200 and 1,600 ft. per min. is permissible. For this purpose velocities are based on an air density corresponding to 70°, this being merely an arbitrary standard adopted for convenience in making computations. In very cold climates a velocity of 800 ft. per min. or less is advisable because of the tendency for the condensation to freeze in the coils. The velocity thus chosen is used both as a basis for computing the height and width of the heater and also for determining its depth, that is, the number of stacks to be used. In Table 76 are given the final temperatures obtainable from heaters of various depths for air at different initial temperatures and velocities. The final temperature for which the heater is designed depends upon the amount of heat to be supplied and upon whether the fan system is to be used for ventilating alone or for supplying the heating requirements also. The temperature of entering air used in the computations should be the minimum for which the system is to be designed.

Example.—Assume that a factory is to be heated and that 1,400,000 cu. ft. of air per hr. is required at 140°. Minimum outside temperature 0°. Ratio volume 140° to 70° air is 1.1322 (from Table 5, Chap. I). What size of Vento heater should be used?

$$\text{Free area (sq. ft.)} = \frac{\text{volume (c.f.m. at } 70^\circ)}{\text{velocity (ft. per min.)}}$$

$$\text{Free area} = \frac{1{,}400{,}000}{1{,}200 \times 60 \times 1.1322} = 17.17 \text{ sq. ft.}$$

Referring to Table 75, it is seen that by using eighteen 60-in. sections, spaced 5 in. center to center, the free area will be 18 × 0.921 = 16.58 sq. ft., which is sufficient, giving a velocity of 1,244 ft. per min. From Table 76 it is seen that a heater seven stacks deep would raise the air from a temperature of 0 to 140° at a velocity of 1,200 ft. per min. The heater should therefore be seven stacks deep. Ordinarily it would be divided into a tempering coil of three stacks and a heating coil of four stacks.

The friction loss through cast-iron heaters is shown in Table 77 for various velocities and various numbers of stacks.

299. Selection of Fin-tube Units.—The heat-transfer characteristics of fin-tube units have been studied and compared in

TABLE 76.—FINAL TEMPERATURES AND CONDENSATION

VENTO HEATERS

(Regular section—standard spacing, 5-in. centers of sections—steam, 227° 5 lb. gage)

Number of stacks deep	Temperature of entering air	Velocity through heater in feet per minute—measured at 70°															
		600		800		1,000		1,200		1,400		1,600		1,800		2,000	
		Final temp. of air leaving heater	Cond. lb. per sq. ft. per hour	F.t.	C.	F.t.	C.	F.t.	C.	F.t.	C.	F.t.	C.	F.t.	C.	F.t.	C.
1	−20																
	−10	34	1.69														
	0	43	1.65	38	1.95	35	2.24	32	2.46								
	20	58	1.46	54	1.75	51	1.99	49	2.23	47	2.42	45	2.50	43	2.65	42	2.82
	30	66	1.39	62	1.64	60	1.92	58	2.17	56	2.33	54	2.40	52	2.54	51	2.69
2	−20	63	1.60	55	1.92	49	2.22	44	2.46	40	2.69	37	2.92	34	3.12	31	3.27
	−10	69	1.52	62	1.85	56	2.12	51	2.35	47	2.56	44	2.77	41	2.94	38	3.08
	0	75	1.44	68	1.74	62	1.99	58	2.23	54	2.42	51	2.62	48	2.77	46	2.95
	20	87	1.29	81	1.57	76	1.80	72	2.00	69	2.20	66	2.36	64	2.54	62	2.69
	30	93	1.21	87	1.46	83	1.70	79	1.89	70	2.06	73	2.21	71	2.37	69	2.50
3	−20	91	1.42	82	1.74	75	2.03	69	2.28	64	2.51	59	2.70	55	2.88	52	3.08
	−10	96	1.36	87	1.66	80	1.92	75	2.18	70	2.39	66	2.60	62	2.77	58	2.90
	0	101	1.30	93	1.59	86	1.84	81	2.08	76	2.27	72	2.46	68	2.62	65	2.78
	20	110	1.15	103	1.42	97	1.65	92	1.85	88	2.06	85	2.22	82	2.38	79	2.52
	30	115	1.09	108	1.33	103	1.56	98	1.75	94	1.91	91	2.08	88	2.23	85	2.35
4	−20	114	1.29	103	1.58	96	1.86	90	2.12	84	2.34	78	2.51	74	2.71	70	2.88
	−10	117	1.22	108	1.51	101	1.78	95	2.02	89	2.22	84	2.41	80	2.60	76	2.76
	0	121	1.16	113	1.45	106	1.70	100	1.92	95	2.13	90	2.31	86	2.48	82	2.63
	20	130	1.06	122	1.31	115	1.52	110	1.73	105	1.91	101	2.08	97	2.22	94	2.37
	30	134	1.00	126	1.23	120	1.44	115	1.63	110	1.80	106	1.95	102	2.08	99	2.21
5	−20	132	1.17	122	1.46	114	1.72	107	1.95	100	2.15	94	2.34	90	2.54	86	2.72
	−10	135	1.13	126	1.40	118	1.64	111	1.86	105	2.06	99	2.24	95	2.42	91	2.59
	0	138	1.06	129	1.32	122	1.56	115	1.77	109	1.96	104	2.14	100	2.31	96	2.46
	20	144	.95	136	1.19	130	1.41	124	1.60	119	1.78	114	1.93	110	2.08	107	2.23
	30	148	.91	140	1.13	134	1.33	128	1.51	123	1.67	118	1.80	115	1.96	112	2.10
6	−20	146	1.06	137	1.34	129	1.59	121	1.81	115	2.02	110	2.22	105	2.40	100	2.56
	−10	149	1.02	140	1.28	132	1.52	125	1.73	119	1.93	114	2.12	109	2.29	104	2.44
	0	152	.97	143	1.22	135	1.44	129	1.65	123	1.84	118	2.02	113	2.17	109	2.33
	20	156	.87	148	1.10	142	1.30	129	1.49	130	1.65	126	1.81	122	1.96	118	2.09
	30	159	.83	151	1.04	145	1.23	139	1.40	134	1.56	130	1.71	126	1.85	122	1.97
7	−20	159	.98	150	1.25	141	1.47	134	1.69	128	1.90	122	2.08	117	2.26	113	2.44
	−10	161	.94	152	1.19	144	1.41	137	1.62	131	1.81	126	1.99	121	2.16	117	2.33
	0	163	.90	154	1.13	147	1.35	140	1.54	135	1.73	130	1.90	125	2.06	121	2.22
	20	167	.81	159	1.02	152	1.21	146	1.39	141	1.55	136	1.70	132	1.85	128	1.98
	30	169	.76	161	.96	155	1.15	149	1.31	144	1.46	139	1.60	135	1.73	132	1.87
8	−20	168	.90	159	1.15	151	1.37	144	1.58	138	1.77	133	1.96	128	2.14	123	2.29
	−10	170	.87	161	1.10	153	1.31	147	1.51	141	1.69	136	1.87	131	2.04	126	2.18
	0	172	.83	164	1.05	156	1.25	150	1.44	144	1.62	139	1.78	134	1.93	129	2.07
	20	175	.75	167	.94	161	1.13	155	1.30	150	1.46	145	1.60	141	1.74	137	1.87
	30	177	.71	169	.89	163	1.07	158	1.23	153	1.38	148	1.51	144	1.64	140	1.76

TABLE 77.—FRICTION OF AIR THROUGH VENTO HEATERS

(Friction loss—in inches of water—due to air passing through Vento stacks (measured at 70°). Regular section—4⅝, 5, and 5⅜-in. spacing.)

Velocity feet per min.	Spacing of sections, inches	1 stack	2 stacks	3 stacks	4 stacks	5 stacks	6 stacks	7 stacks	8 stacks
600	4⅝	0.022	0.043	0.063	0.084	0.105	0.126	0.147	
	5	0.021	0.040	0.058	0.076	0.094	0.112	0.130	0.149
	5⅜	0.019	0.034	0.049	0.064	0.079	0.094	0.109	0.124
700	4⅝	0.031	0.059	0.087	0.115	0.143	0.172	0.200	
	5	0.028	0.054	0.079	0.105	0.130	0.155	0.180	0.205
	5⅜	0.025	0.046	0.066	0.087	0.108	0.128	0.149	0.170
800	4⅝	0.040	0.077	0.114	0.150	0.187	0.224	0.259	
	5	0.037	0.070	0.103	0.135	0.167	0.200	0.232	0.265
	5⅜	0.033	0.060	0.087	0.114	0.140	0.167	0.194	0.221
900	4⅝	0.051	0.097	0.144	0.190	0.237	0.283	0.329	
	5	0.047	0.088	0.129	0.170	0.211	0.252	0.293	0.335
	5⅜	0.042	0.076	0.110	0.144	0.178	0.212	0.246	0.280
1,000	4⅝	0.063	0.120	0.178	0.235	0.293	0.350	0.407	
	5	0.059	0.109	0.160	0.211	0.262	0.313	0.364	0.415
	5⅜	0.052	0.094	0.136	0.178	0.220	0.262	0.304	0.346
1,100	4⅝	0.076	0.145	0.214	0.284	0.353	0.422	0.491	
	5	0.071	0.132	0.193	0.255	0.316	0.377	0.438	0.501
	5⅜	0.062	0.113	0.164	0.215	0.265	0.316	0.367	0.418
1,200	4⅝	0.090	0.172	0.255	0.337	0.420	0.502	0.584	
	5	0.084	0.157	0.230	0.303	0.376	0.449	0.522	0.596
	5⅜	0.074	0.134	0.195	0.255	0.316	0.376	0.437	0.497
1,300	4⅝	0.105	0.202	0.299	0.396	0.493	0.590	0.687	
	5	0.099	0.185	0.271	0.356	0.442	0.528	0.614	0.701
	5⅜	0.087	0.158	0.229	0.300	0.371	0.442	0.513	0.584
1,400	4⅝	0.122	0.234	0.347	0.459	0.572	0.684	0.796	
	5	0.115	0.214	0.314	0.414	0.513	0.612	0.712	0.813
	5⅜	0.101	0.183	0.266	0.348	0.430	0.512	0.595	0.677
1,500	4⅝	0.140	0.269	0.398	0.527	0.656	0.785	0.914	
	5	0.132	0.246	0.360	0.474	0.588	0.702	0.816	0.932
	5⅜	0.116	0.210	0.305	0.399	0.493	0.587	0.682	0.776
1,600	4⅝	0.160	0.306	0.453	0.600	0.746	0.893	1.040	
	5	0.150	0.280	0.410	0.540	0.670	0.800	0.930	1.060
	5⅜	0.132	0.239	0.347	0.454	0.561	0.668	0.776	0.883
1,700	4⅝	0.180	0.346	0.512	0.677	0.843	1.009	1.174	
	5	0.169	0.316	0.463	0.609	0.756	0.903	1.049	1.197
	5⅜	0.149	0.270	0.391	0.512	0.634	0.755	0.876	0.997
1,800	4⅝	0.202	0.387	0.573	0.759	0.944	1.130	1.316	
	5	0.190	0.354	0.518	0.683	0.848	1.012	1.177	1.342
	5⅜	0.167	0.303	0.439	0.575	0.710	0.846	0.982	1.118

the laboratory,[1] but the engineer designing an air-conditioning system usually depends upon the guaranteed performance data of reliable manufacturers.

TABLE 78.—PREFERABLE VELOCITIES THROUGH NET FACE AREAS, AEROFIN HEATERS
(Referred to a temperature of 70°)

Rows of tubes deep	Face velocities, feet per minute	
	Public buildings	Factory buildings
2	600–800	
3	600–700	600–800
4	500–700	600–800
5	500–700	600–800
6	500–600	600–800
7	400–600	600–800
8	400–500	600–700

The desirable velocities, based upon net face areas (rather than free areas) are given in Table 78 for one form of heater.

The selection of the proper depth (that is, number of tubes deep) of the heater is made from Table 79, and the friction drop is obtained from Table 80.

Example.—To heat 14,400 cu. ft. of air per min. from 0 to 80°, using a steam pressure of 5 lb. Assume a face velocity of 600 ft. per min. From Table 79 it will be seen that a unit two rows deep will give a temperature rise from 0 to 88.1°. From Table 80 note that the friction loss is 0.164 in. and the condensation is 2.19 lb. per hr per sq. ft. of surface.

The net face area will be 14,400 ÷ 600 = 24 sq. ft. The proper dimensions would be chosen from the manufacturer's dimension tables.

These heater tables are merely illustrative, and the latest catalogues of the manufacturers should be consulted for the selection of units for an actual building.

The selection of units for cooling involves certain considerations that have not yet been discussed, and the procedure in selecting such units will be deferred to a later chapter. Although in some cases it is possible to use the same units for both cooling

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[1] TUVE and MCKEEMAN, "Heat Transfer from Direct and Extended Surfaces with Forced Circulation," *Trans. A.S.H. & V.E.*, 1934; G. L. TUVE, "Performance of Fin-tube Units for Air Heating," *Trans. A.S.H. & V.E.*, 1936.

TABLE 79.—FINAL TEMPERATURES AND CONDENSATIONS
(5 Lb. Steam 227°, Flexitube Aerofin 71 and 72)

Temp. entering air	Rows tubes deep	Velocity of air through net face area,[1] ft. per min., measured at 70°F. and 29.92 in. barometer											
		300 ft. face velocity		400 ft. face velocity		500 ft. face velocity		600 ft. face velocity		700 ft. face velocity		800 ft. face velocity	
		Final temp. air	Cond. lb. per hr. per sq. ft. h. s.[2]	Final temp. air	Cond. lb. per hr. per sq. ft. h. s.	Final temp. air	Cond. lb. per hr. per sq. ft. h. s.	Final temp. air	Cond. lb. per hr. per sq. ft. h. s.	Final temp. air	Cond. lb. per hr. per sq. ft. h. s.	Final temp. air	Cond. lb. per hr. per sq. ft. h. s.
−20°	1	48.8	1.71	41.7	2.04	37.1	2.36	34.1	2.68	31.5	2.98	27.4	3.13
	2	98.5	1.47	89.5	1.81	81.5	2.10	76.0	2.38	71.8	2.66	66.3	2.85
	3	134.1	1.27	124.6	1.59	116.0	1.87	109.4	2.14	103.7	2.39	97.6	2.59
	4	159.8	1.12	150.8	1.41	142.0	1.67	135.0	1.92	129.0	2.16	122.2	2.35
	5	178.6	0.99	169.8	1.26	162.0	1.51	154.9	1.74	149.1	1.96	142.7	2.15
	6	192.4	0.88	184.5	1.13	177.0	1.36	170.6	1.58	165.0	1.78	158.9	1.97
0°	1	63.2	1.57	56.7	1.87	52.4	2.17	49.7	2.47	47.3	2.74	43.6	2.88
	2	108.9	1.35	100.6	1.66	93.2	1.93	88.1	2.19	84.4	2.44	79.3	2.62
	3	141.6	1.17	132.8	1.46	124.9	1.72	118.9	1.97	113.6	2.19	108.1	2.38
	4	165.2	1.02	156.9	1.30	148.9	1.54	142.4	1.77	136.9	1.98	130.7	2.16
	5	182.5	0.91	174.4	1.15	167.2	1.38	160.7	1.59	155.3	1.80	149.5	1.98
	6	195.2	0.81	187.9	1.04	181.1	1.25	175.2	1.45	170.1	1.64	164.4	1.81
+30°	1	84.8	1.36	79.2	1.63	75.5	1.88	73.1	2.14	71.0	2.37	67.9	2.51
	2	124.6	1.17	117.4	1.45	111.0	1.67	106.5	1.90	103.3	2.12	98.8	2.27
	3	152.9	1.02	145.3	1.27	138.4	1.49	133.2	1.71	128.6	1.90	123.9	2.07
	4	173.4	0.89	166.2	1.13	159.3	1.34	153.6	1.53	148.8	1.72	143.5	1.88
	5	188.5	0.79	181.4	1.00	175.2	1.20	169.5	1.38	164.8	1.56	159.8	1.72
	6	199.4	0.70	193.1	0.90	187.3	1.08	182.0	1.26	177.6	1.42	172.7	1.57

TABLE 79.—FINAL TEMPERATURES AND CONDENSATIONS.—(*Concluded*)

Temp. entering air	Rows tubes deep	Velocity of air through nct face area,[1] ft. per min., measured at 70°F. and 29.92 in. barometter											
		300 ft. face velocity		400 ft. face velocity		500 ft. face velocity		600 ft. face velocity		700 ft. face velocity		800 ft. face velocity	
		Final temp. air	Cond. lb. per hr. per sq. ft. h. s.[2]	Final temp. air	Cond. lb. per hr. per sq. ft. h. s.	Final temp. air	Cond. lb. per hr. per sq. ft. h. s.	Final temp. air	Cond. lb. per hr. per sq. ft. h. s.	Final temp. air	Cond. lb. per hr. per sq. ft. h. s.	Final temp. air	Cond. lb. per hr. per sq. ft. h. s.
+40°	1	92.1	1.29	86.7	1.54	83.2	1.79	81.0	2.03	79.0	2.26	75.9	2.37
	2	129.7	1.11	122.9	1.37	116.8	1.59	111.6	1.78	109.6	2.01	105.3	2.16
	3	156.7	0.97	149.4	1.21	142.9	1.42	137.9	1.62	133.7	1.81	129.1	1.96
	4	176.1	0.84	169.3	1.07	162.6	1.27	157.3	1.46	152.8	1.63	147.7	1.78
	5	190.4	0.75	183.7	0.95	177.8	1.14	172.5	1.31	168.0	1.48	163.1	1.63
	6	200.9	0.66	194.8	0.85	189.3	1.03	184.4	1.19	180.1	1.35	175.5	1.49
+60°	1	106.5	1.15	101.7	1.38	98.6	1.60	96.6	1.82	94.8	2.01	92.1	2.12
	2	140.1	0.99	134.1	1.23	128.6	1.42	124.9	1.61	121.1	1.77	118.4	1.93
	3	164.3	0.86	158.2	1.08	151.9	1.27	147.5	1.45	143.7	1.61	139.6	1.76
	4	181.6	0.75	175.4	0.95	169.5	1.13	164.8	1.30	160.7	1.46	156.2	1.59
	5	194.4	0.67	188.4	0.85	183.0	1.02	178.3	1.17	174.3	1.32	170.0	1.46
	6	203.6	0.59	198.3	0.76	193.3	0.92	189.0	1.07	185.2	1.21	181.0	1.33
+70°	1	113.7	1.08	109.2	1.30	106.3	1.50	104.4	1.71	102.7	1.89	100.2	2.00
	2	145.4	0.94	139.6	1.15	134.5	1.33	131.0	1.51	128.4	1.69	124.9	1.82
	3	168.0	0.81	161.9	1.01	156.4	1.19	152.3	1.36	148.6	1.52	144.8	1.65
	4	184.3	0.71	178.5	0.90	173.0	1.06	168.5	1.22	164.7	1.37	160.4	1.50
	5	196.3	0.63	190.7	0.80	185.7	0.96	181.2	1.10	177.5	1.24	173.4	1.37
	6	205.1	0.56	200.0	0.72	195.3	0.86	191.3	1.00	187.2	1.13	183.8	1.25

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[1] Net face area means only that area facing the tubes and does not include the headers or casings.

[2] Heating surface.

TABLE 80.—FRICTION LOSS THROUGH AEROFIN HEATERS, IN INCHES OF WATER, FOR 70° AND 29.92 IN. BAROMETER (Flexitube 71 and 72)

Rows of helical tubing deep	Velocity of air, ft. per min., through net face area[1]					
	300	400	500	600	700	800
1	0.030	0.050	0.074	0.103	0.136	0.174
2	0.044	0.076	0.116	0.164	0.221	0.285
3	0.058	0.101	0.158	0.225	0.306	0.396
4	0.072	0.127	0.200	0.286	0.391	0.507
5	0.087	0.153	0.241	0.347	0.476	0.618
6	0.101	0.178	0.283	0.408	0.561	0.729
7	0.115	0.204	0.325	0.469	0.646	0.840
8	0.130	0.230	0.367	0.530	0.731	0.951
9	0.144	0.255	0.409	0.591	0.816	1.062
10	0.158	0.281	0.450	0.652	0.901	1.173

[1] Net face area means only that area facing the tubes and does not include headers or casings.

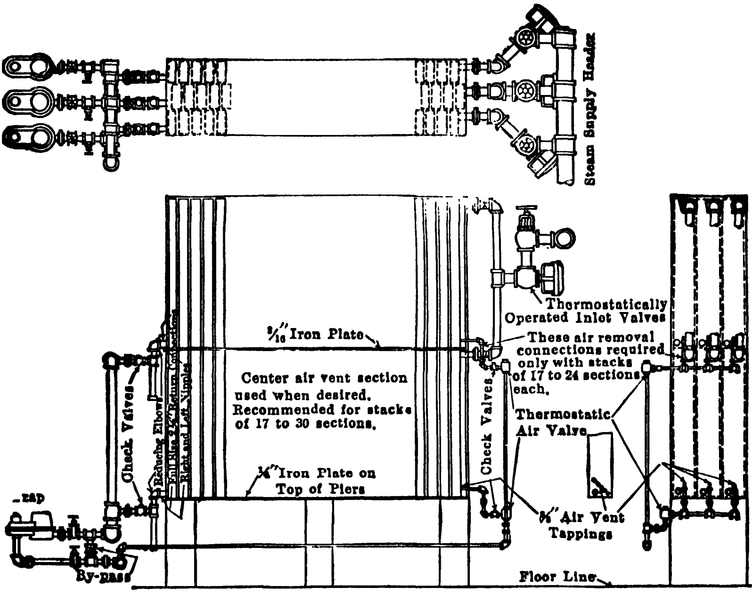
Steam Supply Header
Thermostatically Operated Inlet Valves
These air removal connections required only with stacks of 17 to 24 sections each.
Thermostatic Air Valve
5/16" Iron Plate
Center air vent section used when desired. Recommended for stacks of 17 to 30 sections.
1/4" Iron Plate on Top of Piers
Check Valves
Reducing Elbow
Full Size 2½" Return Connections
Right and Left Nipples
Check Valves
3/4" Air Vent Tappings
-rap
By-pass
Floor Line

FIG. 247.—Piping connections for Vento heaters.

and heating, it is usually preferable to install separate units for each service.

300. Installation and Piping Connections.—Vento heating units are usually mounted on a brick or concrete pier and enclosed by a metal duct. The proper arrangement of the steam piping connections for Vento heaters is shown in Fig. 247 for a double-tier installation. The center section of a long stack is tapped for an air vent as shown. Separate valves should be provided for each stack or pair of stacks.

Special care is necessary in arranging the return connections from fan heaters, as any accumulation of condensation will soon be frozen by the cold air. There is always a considerable drop in pressure through the heaters and the inlet connections, so that the pressure at the return connections should not be depended upon to lift the condensation; the discharge should be by gravity, or a vacuum pump should be used.

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"Size and Characteristics of Air-borne Impurities," by W. G. Frank, *Heating, Piping and Air Conditioning*, January, 1932.

"Air Filter Performance as Affected by Kind of Dust, Rate of Dust Feed and Air Velocity through Filter," by Rowley and Jordan, *Trans. A.S.H. & V.E.*, 1939.

"Comparison of the Weight, Particle Count and Discoloration Methods of Testing Air Filters," by Rowley and Jordan, *Trans. A.S.H. & V.E.*, 1941.

"A New Electrostatic Precipitator," by G. W. Penney, *Elec. Eng.*, January, 1937.

HEATERS

See references accompanying Art. 299.

Manufacturers' catalogues contain valuable installation data.

Problems

1. A fan is delivering 12,000 c.f.m. at a static pressure of 1¼ in., speed 420 r.p.m., and drawing 4 hp. It is desired to increase the fan output to 15,000 c.f.m. What would be the speed, horsepower, and static pressure for the increased output?

2. Find the speed, horsepower, and outlet velocity of a fan that will deliver 20,000 c.f.m. at 120° (the air quantity being given on a 70° basis) against a static pressure of 1⅝ in. Use Table 72.

3. A fan is delivering 23,600 c.f.m. against a static pressure of 1⅞ in. The horsepower is 11.6 and the speed 377 r.p.m. It is desired to increase the delivery to 27,400 c.f.m. through the same piping system. What are the new speed, horsepower, and static pressure?

4. A fan delivers 18,000 c.f.m. against a static pressure of 1½ in. with a speed of 315 r.p.m. and a horsepower of 7.2. The temperature is 35° and the barometric pressure 29.5 in. What would be the speed, static pressure, and horsepower required to deliver 14,300 c.f.m. at a temperature of 115° and with the barometric pressure 30.2 in.?

5. A fan delivers 27,000 c.f.m. through a duct system with a static head of 1.87 in. What head would be required to deliver 32,400 c.f.m.?

6. A fan delivers 34,500 c.f.m. against a static pressure of 2¼ in. through a duct measuring 3 ft. 6 in. by 5 ft. 5 in. The mechanical efficiency of the fan is 71 per cent. What input horsepower is required? Air temperature is 120°, and humidity is 50 per cent.

7. It is desired to carry a relative humidity of 45 per cent at a temperature of 72° in a building. What should be the temperature of the air leaving the air washer?

8. An air washer has a humidifying efficiency of 65 per cent. How many degrees will the incoming air be cooled from an initial temperature of 95° if its wet-bulb temperature is 79°? How much will the relative and absolute humidity be increased?

9. The outside air has a dew point of 62° and a temperature of 80°. After passing through a washer having a humidifying efficiency of 70 per cent, what will be its dew-point and wet-bulb temperatures?

10. It is desired to maintain a dew point of 62° and a temperature of 85° in a closed space. The rate of air change, all supplied by a fan and washer, is 27,500 cu. ft. per hr. (at room conditions). In the room, moisture is being given off from a process at the rate of 0.15 lb. per min. The heat loss from the room is 12,500 B.t.u. per hr., part of which is supplied by steam radiators. The outdoor air is at 50° dry bulb and 42° wet bulb. Calculate (1) the dew-point temperature of the air leaving the washer,

(2) the amount of water added by the washer per hour, (3) the amount of heat added by the washer, and (4) the amount of heat added by the radiators.

11. A room is cooled by evaporative cooling with an air washer having a humidifying efficiency of 95 per cent. The outside conditions are 95° dry bulb and 70° wet bulb. The heat to be absorbed in the room is 72,000 B.t.u. per hr., and the allowable temperature is 83° dry bulb. How much air must be circulated per minute?

12. What would be the size of Vento heater required to heat 800,000 cu. ft. of air per hr. from an outside temperature of 10° to a delivery temperature of 140°? Assume a velocity through the heater of 1,500 ft. per min.

13. What would be the size of Vento heater required to heat 1,100,000 cu. ft. of air per hr. from an outside temperature of 0° to a delivery temperature of 70°? Assume a velocity through the heater of 1,100 ft. per min.

14. Select an Aerofin heater for the conditions given in Prob. 13, assuming a face velocity of 700 ft. per min. What would be the total friction loss? (Steam pressure 5 lb.)

CHAPTER XVII

THE TRANSMISSION AND DISTRIBUTION OF AIR

301. Air Flow in Ducts.—This chapter deals with the design of ducts for the transmission of air and with methods of introducing and distributing air in the rooms.

Air-duct design is a compromise between friction loss on the one hand and excessive duct size on the other. The sizes should be selected so as to give reasonably low friction losses but not so large as to be awkward. The space required for ductwork is often a serious matter architecturally, and of course the cost of sheet-metal work is an important factor.

Because of the uncertainties of fabrication, the selection of air-duct sizes is not an exact process, but it is based on rational principles.

302. Friction Loss.—The general expression for the friction of fluids in pipes [Eq. (3), Chap. IX] is known as the Fanning formula and is approximately applicable to air.

$$P = f\frac{S}{a}D\frac{v^2}{2g} \tag{1}$$

where P = pressure required to overcome friction, lb. per sq. ft.
a = cross-sectional area of duct, sq. ft.
D = density of air, lb. per cu. ft.
v = velocity, ft. per sec.
f = coefficient of friction.
S = area of contact (perimeter × length).

Recent experimental work has led to modifications of the exponents and a determination of values for the coefficient of friction which, when applied to Eq. (1), give the following equation[1] for standard air (70° and 29.92 in. barometer):

$$p = \frac{0.03FL}{d^{1.24}}\left(\frac{V}{1{,}000}\right)^{1.84} \tag{2}$$

---

[1] For derivation and detailed explanation see "Fan Engineering," Buffalo Forge Company (1938 ed., pp. 119*ff*.). Equation (2) is taken from that work.

TABLE 81.—EQUIVALENT ROUND AND RECTANGULAR DUCTS FOR EQUAL FRICTION

Side of rectangular duct, in.	2½	3	3½	4	4½	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19
	Diameter of equivalent round pipe, in.																			
8	4.7	5.2	5.7	6.1	6.5	6.9	7.6	8.2	8.8											
8½	4.9	5.4	5.9	6.3	6.7	7.1	7.8	8.5	9.1	9.6										
9	5.0	5.5	6.0	6.5	6.9	7.3	8.0	8.7	9.3	9.9										
9½	5.1	5.7	6.2	6.6	7.1	7.5	8.2	8.9	9.6	10.2	10.7									
10	5.2	5.8	6.3	6.8	7.2	7.7	8.4	9.2	9.8	10.4	11.0									
10½	5.4	5.9	6.5	7.0	7.4	7.8	8.6	9.4	10.0	10.7	11.3	11.8								
11	5.5	6.0	6.6	7.1	7.6	8.0	8.8	9.6	10.2	10.9	11.5	12.1	12.6							
11½	5.6	6.2	6.7	7.2	7.7	8.2	9.0	9.8	10.5	11.2	11.8	12.4	12.9	13.4						
12	5.7	6.3	6.9	7.4	7.9	8.3	9.2	10.0	10.7	11.4	12.0	12.6	13.2	13.7						
12½	5.8	6.4	7.0	7.5	8.0	8.5	9.4	10.2	10.9	11.6	12.2	12.9	13.5	14.0						
13	5.9	6.5	7.1	7.7	8.2	8.7	9.6	10.4	11.1	11.8	12.5	13.1	13.7	14.3						
13½	6.0	6.6	7.3	7.8	8.3	8.8	9.7	10.6	11.3	12.1	12.7	13.4	14.0	14.6	15.1					
14	6.1	6.7	7.4	7.9	8.5	8.9	9.9	10.8	11.5	12.3	12.9	13.6	14.3	14.9	15.4					
14½	6.2	6.8	7.5	8.1	8.6	9.1	10.1	11.0	11.7	12.5	13.2	13.9	14.5	15.1	15.7	16.2				
15	6.3	6.9	7.6	8.2	8.7	9.2	10.2	11.1	11.9	12.7	13.4	14.1	14.7	15.3	16.0	16.5				
15½	6.4	7.0	7.7	8.3	8.9	9.4	10.4	11.3	12.1	12.9	13.6	14.3	14.9	15.6	16.2	16.8	17.3			
16	...	7.1	7.8	8.4	9.0	9.5	10.5	11.4	12.3	13.1	13.8	14.5	15.2	15.8	16.5	17.1	17.6			
17	...	7.3	8.1	8.6	9.2	9.8	10.8	11.8	12.6	13.5	14.2	15.0	15.7	16.3	17.0	17.6	18.2	18.7		
18	...	...	8.2	8.8	9.5	10.0	11.1	12.1	13.0	13.8	14.6	15.4	16.1	16.8	17.4	18.1	18.7	19.2	19.8	
19	...	...	...	9.1	9.7	10.3	11.4	12.4	13.3	14.2	15.0	15.8	16.5	17.2	17.9	18.6	19.2	19.8	20.4	20.9
20	...	...	...	...	...	10.5	11.6	12.7	13.6	14.5	15.4	16.2	17.0	17.6	18.4	19.0	19.7	20.3	20.9	21.5
21	...	...	...	...	...	10.8	11.9	12.9	13.9	14.9	15.7	16.5	17.4	18.0	18.8	19.5	20.1	20.8	21.4	22.0
22	...	...	...	...	...	11.0	12.1	13.2	14.2	15.2	16.1	16.9	17.8	18.5	19.2	19.9	20.6	21.3	21.9	22.5
23	...	...	...	...	...	11.2	12.4	13.5	14.5	15.5	16.4	17.3	18.1	18.9	19.6	20.4	21.1	21.7	22.4	23.0
24	...	...	...	...	...	11.4	12.6	13.8	14.8	15.8	16.8	17.6	18.5	19.3	20.0	20.8	21.5	22.2	22.8	23.5
25	...	...	...	...	...	11.6	12.8	14.0	15.1	16.1	17.0	17.9	18.8	19.6	20.4	21.2	21.9	22.6	23.3	24.0
26	...	...	...	...	...	11.8	13.1	14.3	15.4	16.4	17.3	18.3	19.2	20.0	20.8	21.6	22.3	23.0	23.8	24.4
27	...	...	...	...	...	12.0	13.3	14.5	15.6	16.7	17.6	18.6	19.5	20.4	21.2	22.0	22.7	23.5	24.2	25.0
28	...	...	...	...	...	12.2	13.5	14.8	15.9	17.0	18.0	19.0	19.8	20.7	21.5	22.4	23.1	23.9	24.6	25.3
29	...	...	...	...	...	12.4	13.7	15.0	16.1	17.2	18.2	19.2	20.2	21.0	21.9	22.7	23.5	24.3	25.0	25.7

30	...	...	...	...	...	12.6	13.9	15.2	16.4	17.5	18.5	19.5	20.5	21.4	22.2	23.1	23.9	24.7	25.4	26.2
31	...	...	...	...	...	12.7	14.1	15.4	16.6	17.7	18.8	19.8	20.8	21.7	22.5	23.4	24.2	25.0	25.8	26.6
32	...	...	...	...	...	12.9	14.3	15.6	16.9	18.0	19.1	20.1	21.1	22.0	22.9	23.8	24.6	25.4	26.2	27.0
33	...	...	...	...	...	13.1	14.5	15.8	17.1	18.2	19.3	20.4	21.4	22.3	23.2	24.1	25.0	25.8	26.5	27.3
34	...	...	...	...	...	13.2	14.7	16.1	17.3	18.5	19.6	20.7	21.6	22.6	23.5	24.4	25.3	26.2	26.9	27.7
35	...	...	...	...	...	13.4	14.9	16.2	17.5	18.7	19.8	20.9	21.9	22.9	23.9	24.7	25.6	26.5	27.2	28.1
36	...	...	...	...	...	....	15.1	16.4	17.7	19.0	20.1	21.2	22.2	23.2	24.2	25.1	26.0	26.8	27.7	28.5
37	...	...	...	...	...	....	15.2	16.6	17.9	19.2	20.3	21.4	22.5	23.5	24.5	25.4	26.3	27.1	28.0	28.8
38	...	...	...	...	...	....	15.4	16.8	18.2	19.4	20.6	21.7	22.8	23.8	24.8	25.8	26.7	27.5	28.4	29.2
39	...	...	...	...	...	....	15.5	17.0	18.4	19.6	20.8	21.9	23.0	24.1	25.1	26.1	27.0	27.8	28.7	29.5
40	...	...	...	...	...	....	15.7	17.2	18.6	19.8	21.1	22.2	23.3	24.4	25.4	26.4	27.3	28.2	29.1	29.9
41	...	...	...	...	...	....	15.9	17.4	18.8	20.0	21.3	22.4	23.5	24.6	25.6	26.6	27.6	28.5	29.4	30.3
42	...	...	...	...	...	....	16.1	17.6	19.0	20.3	21.6	22.7	23.8	24.9	25.9	26.9	27.9	28.8	29.8	30.7
43	...	...	...	...	...	....	....	17.8	19.2	20.5	21.8	22.9	24.0	25.1	26.2	27.2	28.2	29.1	30.0	30.9
44	...	...	...	...	...	....	....	18.0	19.4	20.7	22.0	23.1	24.3	25.4	26.5	27.5	28.5	29.5	30.3	31.2
45	...	...	...	...	...	....	....	18.2	19.6	20.9	22.2	23.3	24.5	25.6	26.7	27.8	28.8	29.8	30.6	31.5
46	...	...	...	...	...	....	....	18.4	19.8	21.1	22.4	23.6	24.8	25.9	27.0	28.1	29.1	30.1	31.0	31.9
47	...	...	...	...	...	....	....	18.6	19.9	21.3	22.6	23.8	25.0	26.1	27.2	28.3	29.3	30.3	31.3	32.2
48	...	...	...	...	...	....	....	....	20.1	21.5	22.8	24.1	25.2	26.4	27.5	28.6	29.6	30.5	31.6	32.5
49	...	...	...	...	...	....	....	....	20.2	21.7	23.0	24.3	25.4	26.6	27.7	28.9	29.9	30.9	31.9	32.8
50	...	...	...	...	...	....	....	....	20.4	21.9	23.2	24.5	25.7	26.9	28.0	29.2	30.3	31.3	32.2	33.1
51	...	...	...	...	...	....	....	....	20.6	22.0	23.4	24.7	25.9	27.1	28.2	29.4	30.5	31.5	32.5	33.4
52	...	...	...	...	...	....	....	....	20.8	22.2	23.6	24.9	26.2	27.4	28.5	29.6	30.7	31.8	32.9	33.8
53	...	...	...	...	...	....	....	....	20.9	22.4	23.8	25.1	26.4	27.6	28.7	29.8	30.9	32.0	33.1	34.1
54	...	...	...	...	...	....	....	....	21.1	22.6	24.0	25.3	26.6	27.8	29.0	30.1	31.2	32.3	33.4	34.4
55	...	...	...	...	...	....	....	....	21.3	22.7	24.2	25.5	26.8	28.0	29.2	30.3	31.4	32.5	33.6	34.6

Calculated from the formula $D = 1.265\sqrt[5]{\frac{(AB)^3}{A + B}}$,
in which A and B = sides of rectangular duct.
D = diameter of round pipe.

where F = factor for roughness.
L = length of duct, ft.
d = diameter of duct, in.
p = pressure loss, in. of water.
V = velocity, ft. per min.

For average ducts such as are used in air-conditioning work the proper value of $F = 1$.

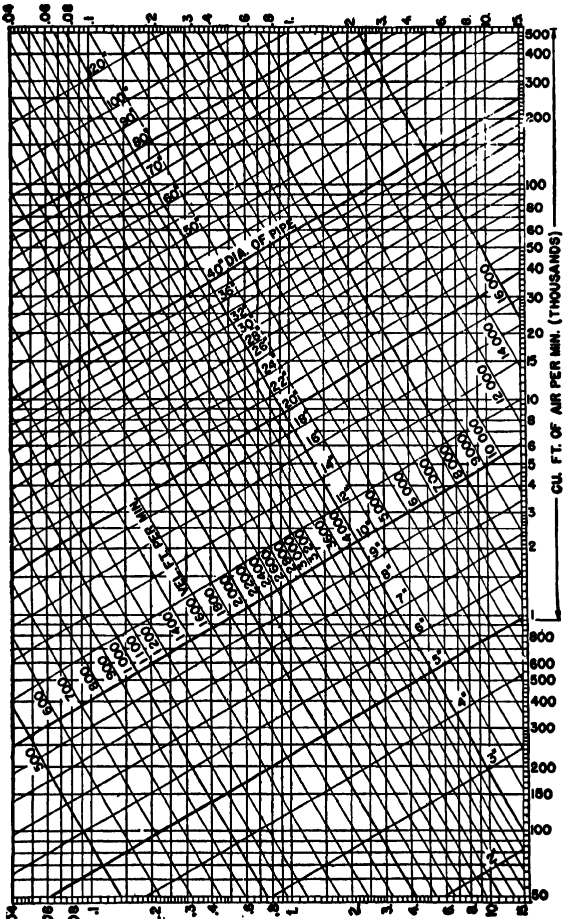

FIG. 248.—Friction loss in round air ducts. (*From "Fan Engineering," Buffalo Forge Company; reproduced by permission.*)

The chart in Fig. 248 is based on Eq. (2) and is appropriate for average ducts. For perfectly smooth ducts, values for friction loss equal to 80 per cent of the chart values are recommended; and for rough conduits of concrete or riveted pipe, 120 per cent of the chart values should be used.

As an example of the use of the chart, assume that 10,000 c.f.m. are passed through 75 ft. of 24-in.-diameter pipe. Select 10,000 c.f.m. on the right scale, and move horizontally left to the diagonal line marked 24 in. The other intersecting diagonal shows that the velocity in the pipe is 3,200 ft. per min. Directly below the intersection the friction per 100 ft. is found to be 0.49 in.; then for 75 ft. it will be 0.75 × 0.49 = 0.37 in. In a similar manner any two variables may be determined by the intersection of the lines representing the other two variables.

303. Correction for Temperature.—The chart in Fig. 248 is for standard air (70° and 29.92 in. barometer). From Eq. (1) it is seen that the friction loss varies directly as the density and therefore varies directly as the barometric pressure and inversely as the absolute temperature. Thus:

$$\begin{aligned} P_{\text{actual}} &= P_{\text{chart}} \times \frac{B}{29.92} \times \frac{459.7 + 70}{459.7 + t} \\ &= P_{\text{chart}} \times \frac{17.70B}{459.7 + t} \qquad (3) \end{aligned}$$

where B = barometric pressure, in. Hq.
t = temperature, deg. F.

The correction for barometric pressure is necessary only for installations that are much above or below sea level. The temperature correction should be used with discretion, because in many systems the air temperature varies, and the temperature for which it is desired to design the system should be selected.

It should be noted particularly that the air quantity or velocity to be used in making use of the friction chart or of Eq. (2) is the actual quantity or actual velocity at the stated temperature. Therefore if the air quantity is given on a 70° basis, as is often the case, it must be converted to actual volume or velocity before the chart or formula is used.

304. Friction Loss in Rectangular Pipes.—The friction chart is for round pipes. A rectangular pipe that is equivalent to a given round pipe is one that will have the same carrying capacity and pressure drop. Table 81 gives the diameters of round pipes equivalent to rectangular pipes of stated dimensions.

305. Pressure Loss Due to Obstructions.—The loss of pressure caused by various obstructions, such as elbows and branches, is usually expressed as a multiple of the velocity head. The actual loss, however, is of course a loss of static head, since the velocity head at all points in a pipe, for a given quantity of air flowing, depends entirely upon the cross-sectional area at each point.

TABLE 82.—PRESSURE LOSS DUE TO ELBOWS

Center-line radius, per cent of pipe width	Velocity head lost, per cent	
	Round duct	Square duct
50	75	105
75	38	37
100	26	21
150	17	13
200	15	11
Sharp corner, no curves or vanes	87	125

Table 82 shows the loss for elbows of various center-line radii, with the radius expressed in percentage of the pipe width in the plane of the elbow. It is evident from Table 82 that an appreciable reduction in pressure loss can be accomplished by making the radius 1½ times the pipe width, which means that the inner radius is equal to the width. Furthermore, there is little to be gained by further increasing the radius.

Where a square-corner elbow must be used, the addition of turning vanes as shown in Fig. 249 is a marked improvement. If the design is carefully worked out, the pressure loss is reduced to as little as one-fourth of the velocity head.[1] The reason for this is that the vanes split the air flow into a series of smooth streams.

---

[1] LORING WIRT, "New Data for the Design of Elbows in Duct Systems," *Gen. Elec. Rev.*, June, 1927. See also STUART, WARNER, and ROBERTS, "Effect of Vanes in Reducing Pressure Loss in Elbows in 7-Inch Square Ventilating Duct," *Trans. A.S.H. & V.E.*, 1942.

Figure 249 shows the correct proportions for a duct of 12 in. width.[1] Sets of turning vanes for insertion into ductwork are available commercially.

Still further reduction in velocity pressure may be obtained by thickening the shape of the vane as shown dotted in Fig. 249, so that the area at the corner and consequent velocity are uniform. With this arrangement the velocity-head loss may be reduced to about 10 per cent. For sharp turns, particularly where an inside radius cannot be used, these turning vanes have found useful application.

In round-corner elbows in which the center line is less than 150 per cent of the pipe width, the loss is appreciably reduced by installing two or three vanes or splitters concentric with the radii of the duct walls. The effect is to divide the elbow into three or four elbows having a more favorable ratio of center-line radius to width, with the consequent improvement indicated by the values in Table 82.[2]

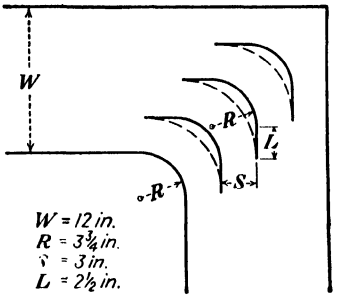

FIG. 249.—Square-corner elbow with turning vanes.

The data in Table 82 are for pipes of approximately square or round cross section. The pressure loss is affected by the radius of curvature and, in rectangular ducts, by the ratio of the width to the depth of the cross section. These rather complex relationships require a detailed study if extreme accuracy is called for. For general work the values in Table 82 may be used without serious error, but where a duct system is extensive it may be profitable to study the question of elbow loss in detail.

The following table gives the loss of pressure due to various obstructions.

Example.—Given an air duct of square cross section carrying air at a velocity of 900 ft. per min. and at a temperature of 70°. Find the loss

---

[1] From Wirt data. See "Air Conditioning and Engineering," American Blower Corp., for chart showing proportions for various sizes of duct.

[2] For a full discussion of elbow design see Madison and Parker, "Pressure Losses in Rectangular Elbows," *Heating, Piping and Air Conditioning*, July, August, and September, 1936. See also "Fan Engineering," Buffalo Forge Company.

TABLE 83.—PRESSURE LOSS DUE TO VARIOUS OBSTRUCTIONS

	Loss, Per Cent of Velocity Head
Round corner, square duct, center-line radius 1½ × depth	13
Round corner, round duct, center-line radius 1½ × diam	17
Square tee	100
Branch from main duct	
Angle, 15 deg. (per cent velocity pressure in branch)	10
30 deg	20
45 deg	25
Abrupt entrance to pipe	50 to 90
Coned entrance to pipe	25
Registers (free area = duct area = ½ total area of register)	125

Air Washers:

Velocity through Washer, Ft. per Min.	Pressure Loss, In. Water
400	0.15
500	0.25
600	0.35
700	0.45

of head due to an elbow of radius 1½ × width. From Eq. (2), Chap. XVI, the velocity head $= \left(\frac{900}{1{,}096.5}\right)^2 \times 0.07495 = 0.0505$ in. The pressure loss is $0.13 \times 0.0505 = 0.006$ in.

306. Proportioning Duct Systems.—It is desirable that the size of the ducts be intelligently selected and that the pressure loss in the system be computed as accurately as possible. The principal reason for doing this is to ensure the selection of a fan of the proper characteristics; for in order that the required quantity of air may be delivered, it is necessary that a fan be selected with working head sufficient to overcome the resistance of the system. Furthermore, the proper proportioning of the various branches will result in the delivery of the proper air quantities to the various rooms without too great a dependence upon the use of the dampers.

It is not to be assumed, however, that extremely accurate calculations of friction losses can be made in advance of the ductwork construction. Imperfect workmanship and the inevitable variations required to meet field conditions make it unwise to place too great dependence upon the calculations; and a reasonable margin in fan head and other critical factors should be allowed.

In designing a duct system the location of the fan and other equipment is first spotted, and the location and size of the grilles, with the air quantities for each, are determined. The general type of system is decided upon, and the approximate velocities are selected. The duct sizes are then calculated.

There are two general systems of air distribution, and the method of choosing the duct sizes depends upon the type of system. In some cases the individual-duct system described in Chap. XV is used, in which the air is delivered to a plenum chamber by the fan and separate ducts radiate to the various rooms. In such a system the duct having the greatest resistance is first designed, which fixes the pressure to be carried in the plenum chamber. The other ducts are then so designed as to deliver the required quantities with the given pressure differential.

In many systems the trunk duct is used, consisting of one or more main ducts with branches taken off at intervals. Such ducts are usually so proportioned as to give an equal friction loss per foot of length. The outlets are designed for certain velocities depending upon the size of the room and upon the distance through which it is desired to blow the air. The branches from the main duct should be so proportioned as to deliver the required air quantities, and it is usually best to provide dampers on the outlets so that any inequalities in distribution can be adjusted after the system is installed. It is desirable to design all air ducts on a basis of the maximum air density to be expected.

Because of the necessity of avoiding noise, air velocities in public buildings, office buildings, and schools are lower than those

TABLE 84.—VELOCITIES IN DUCTS WHERE QUIET OPERATION IS NECESSARY

	Feet per Minute
Air intakes and connections to and from heaters	1,000
Individual-duct systems:	
Horizontal ducts	700 to 1,200
Vertical ducts, approaching grilles	400 to 750
Wall registers[1]	200 to 400
Floor registers[1]	125 to 175
Trunk-duct systems:	
Main duct	900 to 1,200
Branch ducts	600 to 1,000
Registers or grilles[1]	200 to 400

[1] Over gross area.

used in industrial work. Velocities of 900 to 1,200 ft. per min. represent good practice. Table 84 gives suitable velocities at various points in the duct system.

The over-all static head against which the fan must operate usually works out about as follows:

	In.
School system:	
Plenum-chamber	1¼
Individual-duct	1½
Theaters with artificial cooling	1¼ to 1½
Small summer cooling systems	1 to 1¼
Systems for industrial buildings	1 to 1½

307. Example of Air-duct Design.—The procedure in selecting the air-duct sizes for a trunk duct is illustrated in the following typical problem.

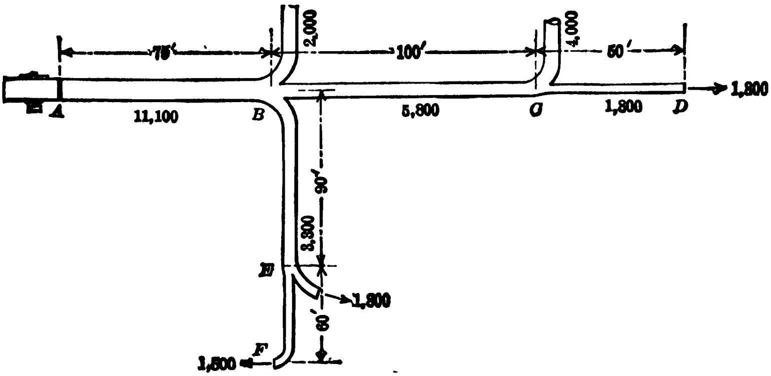

FIG. 250.

Assume a system laid out as in Fig. 250. The quantities are given in the layout as calculated on a basis of 70°. The system, however, is to be designed for a temperature of 135°, and the *actual* quantities flowing in the various sections are therefore as follows:

AB 11,100 × 1.1230[1] = 12,465 c.f.m.
BC 5,800 × 1.1230 = 6,513 c.f.m.
CD 1,800 × 1.1230 = 2,021 c.f.m.
BE 3,300 × 1.1230 = 3,706 c.f.m.
EF 1,500 × 1.1230 = 1,684 c.f.m.

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[1] Conversion factor 1.1230 may be established by taking ratio of absolute temperatures (459.7 + 135) ÷ (459.7 + 70) = 1.1230 or by interpolation from Table 63, under column headed, "Ratio vol. to vol., at 70°F."

The total head at point *A* must be equal to the friction loss in the trunk duct plus the velocity head at *D*, the end of the trunk duct, plus the static pressure that it is desired to carry at *D*. (Each outlet will have a damper by means of which this static pressure can be adjusted.)

The method of proportioning by a uniform friction loss leads to a reduction in the velocity toward the end of the trunk and a consequent conversion of some of the velocity head to static head. The absolute values of the velocity and static heads at *A* are therefore not important, the requirement being that their sum be sufficient. Assuming a velocity of 1,000 ft. per min., the velocity head at *D* will be equal to $\left(\frac{1{,}000}{1{,}096.5}\right)^2 \times 0.06675 = 0.055$ in. on a basis of 135°. A static pressure of 0.25 in. at point *D* will be ample to give the desired velocity and leave a margin for damper adjustment. The friction drop may be fixed arbitrarily, and in this case will be chosen as 0.20 in. per 100 ft., giving a static pressure at point *A* of $0.20 \times 2.25 + 0.25 = 0.70$ in.

The friction loss for 135° air is equal to the loss with 70° air multiplied by the ratio $\frac{459.7 + 70}{459.7 + 135} = 0.89$. For an actual loss of 0.20 in. per 100 ft. assume, in using the friction chart, a loss of $0.20 \times \frac{1}{0.89} = 0.225$ in. For a friction drop of 0.225 in. per 100 ft. the diameters of sections *AB*, *BC*, and *CD* would be, respectively, 30, 24, and 16 in. The diameter of the outlet at *D* would be increased to 19 in. to give the required outlet velocity of 1,000 ft. per min. The branch pipe could be designed for the same pressure loss per unit length, but it is more economical to take advantage of the full available head and reduce the size of the pipe. The static head at *B* can be found by subtracting from the static head at *A* the loss in section *AB*. Allowing for the loss due to entrance in the branch at *B* and for the final velocity and static heads at *F*, the allowable friction loss in sections *BE* and *EF* can be determined and the size of pipe chosen accordingly.

308. Example of Individual-duct System.—Assume that an individual-duct system is to be designed and that the longest duct is arranged as in Fig 251. The air volume when corrected for the actual temperature is 1,600 c.f.m., the temperature being 120°.

The horizontal run will be figured on a basis of 1,000 ft. per min., and a duct of rectangular section will be used. The area of the horizontal duct will be 1,600 ÷ 1,000 = 1.6 sq. ft., and a 12- by 19-in. duct will be used. For the riser a velocity of 750 ft. per min. will be used, and the required area is 1,600 ÷ 750 = 2.13 sq. ft., requiring a 16- by 19-in. duct. From Table 81, the diameter of a round pipe equivalent to a 12- by 19-in. rectangular duct is 16.5 in., and for a 16- by 19-in. duct 19.2 in. From the chart in Fig. 248 it will be found that a pipe of 16.5-in. diameter will transmit 1,600 c.f.m. with a friction loss of 0.10 in. per 100 ft. and that the loss for a 19.2-in. pipe is 0.05 in. per 100 ft.

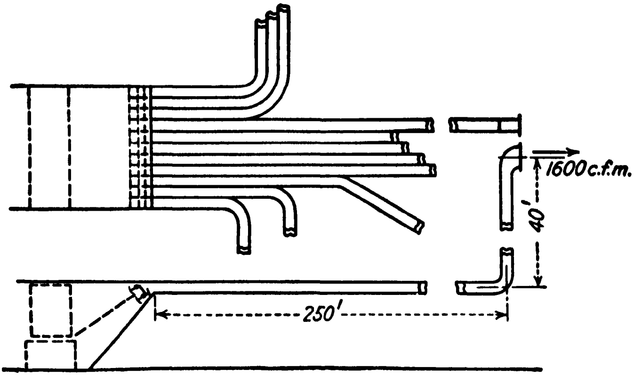

Fig. 251.

The total friction loss for the straight pipe at 70° is then $250 \times \frac{0.10}{100} + 40 \times \frac{0.05}{100} = 0.27$. For 120° air it is $0.27 \times \frac{459.7 + 70}{459.7 + 120} = 0.25$. Elbows will be used with a ratio of radius to width of 1.5, giving a loss of approximately 13 per cent of the velocity head (Table 82).

The velocity heads are calculated by Eq. (2), Chap. XVI.

$$h = \left(\frac{1,000}{1,096.5}\right)^2 \times 0.068 = 0.056$$

$$h_v = \left(\frac{750}{1,096.5}\right)^2 \times 0.068 = 0.032$$

The elbow loss is $0.056 \times 0.13 + 0.032 \times 0.13 = 0.011$ in.

The resistance of the register may be taken as 1.25 times the velocity head corresponding to a register velocity of 300 ft. per min. The velocity head at the register is

$$h_v = \left(\frac{300}{1{,}096.5}\right)^2 \times 0.068 = 0.005 \text{ in.}$$

The loss through the register is 0.005 × 1.25 = 0.006 in. The loss at the entrance to the duct from the plenum chamber may be taken as 80 per cent of the velocity head corresponding to the velocity of 1,000 ft. per min.

$$0.80 \times h_v = 0.80 \times \left(\frac{1{,}000}{1{,}096.5}\right)^2 \times 0.068 = 0.045 \text{ in.}$$

The total resistance of the duct is then

$$0.25 + 0.01 + 0.006 + 0.045 = 0.31 \text{ in.}$$

The total pressure in the plenum chamber should be equal to this plus whatever static pressure it is desired to carry at the ends of the ducts. A static of about 0.10 to 0.15 in. is desirable to make the system stable in operation and to keep it from being affected by the opening of windows or doors. The volume damper in each duct is adjusted to maintain the desired static unless the type of register or grille selected has a high resistance.

The remaining ducts should be designed to use up the available head of 0.31 in. Assume the following data for one of the ducts:

Quantity of air at 120°	1,150 c.f.m.
Total length of pipe	110 ft.
Elbows	3.
Register velocity	300 ft. per min.

The average friction loss per 100 ft. = 0.31/1.1 = 0.28 in. approximately. A 13-in. round pipe would carry this quantity with a friction loss of 0.20 in. per 100 ft. for 70° air. An equivalent rectangular duct would be 10 by 14 in. For air at 120° the loss would be

$$0.20 \times \frac{529.7}{579.7} = 0.183 \text{ in. per 100 ft.}$$

For the entire length the loss would be $0.183 \times {}^{110}\!/_{100} = 0.20$ in. The velocity would be 1,180 ft. per min., and the velocity head would be = $(1{,}180/1{,}096.5)^2 \times 0.068 = 0.079$ in. The elbow resistance would be

$$3 \times 0.13 \times 0.079 = 0.03 \text{ in.}$$

The velocity head required for the register would be, as before, 0.005 in. The entrance loss at the plenum chamber would be $0.80 \times 0.079 = 0.60$.

The total friction loss would be $0.20 + 0.03 + 0.06 + 0.006 = 0.296$, which is sufficiently close to the desired figure of 0.31. The velocity of the riser portion of the duct should preferably be reduced to 750 ft. per min. or thereabouts, to reduce noise and give good approach conditions, which would reduce the friction loss further.

The foregoing calculation has been carried through in a more detailed fashion than would be done on most installations in practice. The fact that the final adjustments are invariably made with dampers renders it useless to calculate losses less than 0.01 in.

309. Construction of Air Ducts.—Air ducts are ordinarily built of galvanized steel, the sections being joined by a lock joint that prevents air leakage. The proper weight of metal for ducts of various sizes is shown in Table 85.

TABLE 85.—RECOMMENDED WEIGHT OF METAL FOR AIR DUCTS

Round ducts, diameter, in.	Rectangular ducts, width, in.	Thickness of metal, U. S. Standard gauge number
6 to 19	4 to 18	26
20 to 29	19 to 30	24
30 to 39	31 to 60	22
40 to 49	61 to 118	20
50 and over	120 and over	18

The more common forms of sheet-metal joints used in ductwork are illustrated in Fig. 252.

At *a* is shown the so-called "Pittsburgh lock joint," which is widely used as a longitudinal corner seam. The joint is closed by hammering over the projecting edge.

Joint *b* is sometimes used as an alternate to *a*, particularly when the inside corner of the duct must be square. This joint requires a dolly bar inside when being hammered closed.

Joint *c* is a riveted joint used for heavy gages (18 and 16).

Joint *d* is used for branch connections.

Joint *e* is used for girth joints along with joint *f*. Two opposite sides of the duct are joined with one method, and the other two sides by the other method.

Joint *g* is used to attach a cap to the end of a duct.

Joints *h*, *i*, and *j* are standing seams, used to stiffen the flat sides of the ducts over 24 in. wide.

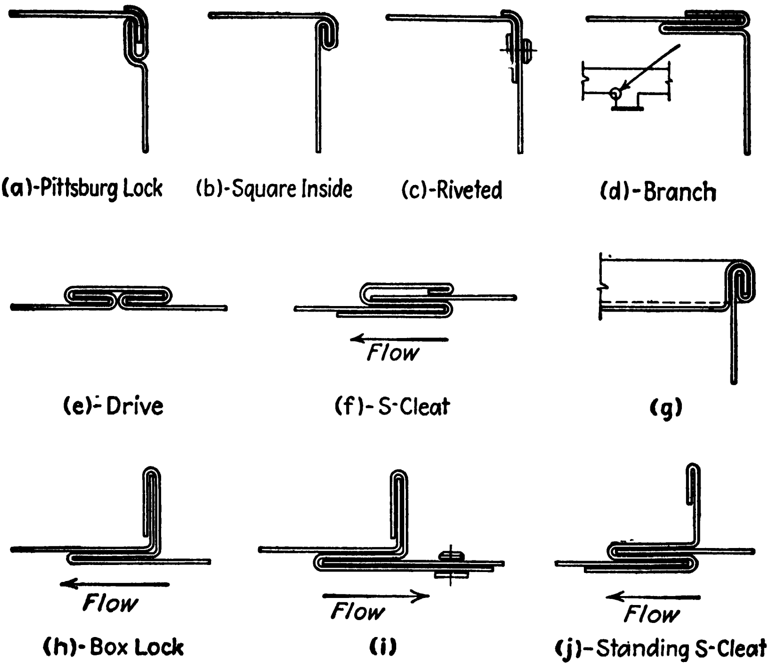

FIG. 252.—Sheet-metal joints used in ductwork.

The ducts should be designed to produce smooth streamlines and without sharp turns or sudden changes of section. Transformation pieces, which change the shape or size of the duct, should have a slope of not more than 1 in 10 in. on each side if possible.

Rectangular ducts should be as nearly square as possible, and in no case should the ratio of the sides be greater than 8:1.

Volume dampers, readily accessible and capable of being locked in position, should be installed in all branch ducts or outlets.

Masonry ducts, such as the plenum chambers in some types of school systems, must be tightly constructed to avoid leakage, and special supervision on this point during construction is necessary.

Insulation of warm-air ducts to avoid loss of heat and to avoid heating of the spaces through which they run is often quite necessary. When the warm-air and cool-air sections of a plenum chamber are adjacent, they must be separated by insulation.

The **heat loss** from warm-air ducts or the heat absorbed by cold-air ducts can be readily calculated by means of the equations and data in Chap. II. The resistance to heat flow of the metal in the duct wall is negligible, and the surface resistances control the heat flow. If the air velocity in the duct is 1,000 ft. per min. (11.4 m.p.h.), the surface conductance for the inside of the duct is found from Fig. 8, Chap. II, to be 4.2; and for the outside of the duct (still air) it is 1.5. For an uninsulated duct the heat-transfer coefficient is therefore

$$U = \frac{1}{\frac{1}{4.2} + \frac{1}{1.5}} = 1.1 \text{ B.t.u. per hr. per deg. difference}$$

If the duct is insulated with a material whose specific conductivity is 0.30 and whose thickness is ¾ in., the heat-transfer coefficient becomes

$$U = \frac{1}{\frac{1}{4.2} + \frac{0.75}{0.30} + \frac{1}{1.5}} = 0.29$$

The insulation may be placed either on the outside or, if the duct is large, on the inside. In the latter case the material must be noncombustible. Some noncombustible insulating materials are also sound absorbing and are therefore useful on the inside surfaces.

Leakage from ducts is not always negligible. If the duct runs through the heated or cooled space, there is no actual loss; but if it does not, the loss may be from 5 to 15 per cent as measured between the fan and the supply grilles.[1]

310. Noise Control.—Increasing noise consciousness and the requirements of new arts such as radio broadcasting and sound moving pictures have made it necessary to study and reduce the noise from air-conditioning apparatus.

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[1] See "Temperature Drop in Ducts for Forced-air Heating Systems," *Univ. Ill. Eng. Exp. Sta. Bull.* 351.

In a central-fan system noisiness is caused by (*a*) the transmission of vibration from fans and motors through the building structure, (*b*) the transmission of fan and duct noise through the ducts, and (*c*) the interference by grilles at the air inlets.

Fan and motor vibration can be isolated from the building structure by mounting the apparatus on supports having pads of cork, rubber, or felt, or on steel springs. There are commercial designs of such structures that are satisfactory if properly selected, and they should be used if there is any probability of trouble. Sound-insulating partitions should be used to enclose a fan room if it is located near other rooms that must be quiet.

Noise that is transmitted through ducts may be caused by the fan blades as they pass the lip of the fan casing or by the high-velocity flow of the air through the ducts. An elementary requirement is a canvas connection between the fan and the duct to stop direct metallic transmission. Another important point is to limit the tip speed of the fan and the outlet velocity. The tip or peripheral speed for a given static pressure depends somewhat upon the design of the fan. For a multiblade fan with forward-curved blades operating against a static pressure of 1 in., a tip speed of 3,000 ft. per min. and an outlet velocity of 1,700 ft. per min. would be good practice. However, if extreme quietness is essential, a larger fan operating at a lower speed would be chosen. Noise is often caused by driving too small a fan at excessive speeds.

The duct velocities given in Table 84 result in fairly quiet operation. Abrupt changes in duct cross section should be avoided, and loose dampers or partitions that may vibrate must be made tight. Where extreme quietness is essential the ducts may be lined with sound-absorbing material such as felt or fiber sheets. The authors have found that good results are obtained in such places as the inlets to auditorium plenum chambers by the use of fiberboard arranged in egg-crate or honeycomb fashion. All such materials should be noncombustible.

The noise caused by grilles can be eliminated only by using low outlet velocities or by avoiding complicated shapes with sharp edges if the velocity cannot be reduced.

311. Noise Measurement.—It is only recently that methods and instruments have been designed for the practical measurement of noise, but surely in the future definite noise specifications will be written for apparatus and for air-conditioning systems.

The unit of noise measurement is the decibel (db), which is not a physical quantity but a ratio. Research has shown that the sensation response due to a change of sound intensity from I_a to I_b is proportional to log (I_b/I_a). The decibel is derived from this relationship. Thus:

$$\text{Sound level (db)} = 10 \log_{10} \frac{I_1}{I_0} \qquad (4)$$

where I_1 = intensity of noise.

I_0 = intensity of noise at reference level.

Unfortunately the value for I_0 is not completely standardized, but the value adopted by the A.S.A. of 1×10^{-16} watt per square centimeter is becoming generally used. The foregoing mathematical relationship corresponds quite well with the actual perception of noise by the human ear. One decibel is about the smallest change in sound level that can be perceived by the human ear. An important point to remember about the decibel unit is that sensation levels expressed in decibels cannot be added; only the *intensities* can be added.

A simple relation to remember is that doubling the sound power is about equal to an increase of 3 decibels, so that when equal sound levels are added together, regardless of their actual value, the resulting level is 3 decibels higher than that of the originals. Thus, two sounds each having a level of 40 db would produce a level of 43 db when combined. Take, for example, a fan which has a noise level of 43 db and assume that it is operating in a room having originally a noise level of 35 db. What is the combined noise level?

From Eq. (4),

$$35 = 10 \log \frac{I_1}{I_0} \qquad (5)$$

and

$$43 = 10 \log \frac{I_f}{I_0} \qquad (6)$$

where I_1 and I_f are the intensities in the room originally and of the fan, respectively.

From Eq. (5),

$$\begin{aligned} 3.5 &= \log \frac{I_1}{I_0} \\ 3.5 &= \log I_1 - \log 1 \times 10^{-16} \\ \log I_1 &= 3.5 + \log 1 \times 10^{-16} \\ \log I_1 &= 3.5 + (0.0000 - 15)\ * \\ \log I_1 &= 0.0000 - 11.5 \end{aligned}$$

Adding the quantity 0.5 to the mantissa and subtracting it from the characteristic,

$$\begin{aligned} \log I_1 &= 0.5000 - 12 \\ I_1 &= 3.16 \times 10^{-13}\ \dagger \end{aligned}$$

Similarly, from Eq. (6),

$$\begin{aligned} \log I_f &= 4.3 + \log 1 \times 10^{-16} \\ &= 4.3 + (0.0000 - 15) \\ &= 0.0000 - 10.7 \\ &= 0.3000 - 11 \\ I_f &= 2.0 \times 10^{-12} \\ &= 20 \times 10^{-13} \\ I_1 + I_f &= 23.16 \times 10^{-13} \\ db &= 10 \log \frac{23.16 \times 10^{-13}}{1 \times 10^{-16}} \\ db &= 10 \log 23.16 \times 10^{3} \\ &= 10(0.3647 + 4) \\ db &= 43.65, \text{ or nearly } 44 \end{aligned}$$

Thus the combined effect of a sound level of 35-db room noise and 43-db fan noise is 44 db.

Sound is measured and the sound level is given in decibels by an instrument called the acoustimeter. The noise produced by a piece of apparatus and the noise level existing in a room can thus be readily measured. It is necessary in the case of machinery noise to specify exactly where the pickup microphone of the acoustimeter is to be located with respect to the machine.

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* The logarithm, as the reader will recall, consists of the mantissa, which is obtained from the table of logarithms, and the characteristic, which corresponds to the number of places to the right of the decimal point less 1. Thus the mantissa for 1×10^{-16} is 0.0000 and the characteristic is 15, since 1×10^{-16} is "decimal point, fifteen zeros, one."

† Similarly, the exponent of 10 is the characteristic *increased* by 1.

A standard test procedure[1] has been formulated for measuring the sound produced by ventilating fans. The fan being tested is provided with a standard length of outlet duct, and seven sound readings are taken around the fan at a distance equivalent to one wheel diameter, but not less than 5 ft. from the edge of the housing. Authorities agree that where the variation of the seven readings is less than 12 db, the arithmetic average is a sufficiently close approximation. A correction for variation in fan speed may be made in accordance with the following formula:

$$\text{Db change} = 50 \log_{10} \frac{\text{RPM}_2}{\text{RPM}_1} \tag{7}$$

Example.—Suppose a fan operating at 600 r.p.m. has a noise rating of 40 db and it is desired to estimate the sound rating of this fan when operating at a speed of 660 r.p.m. Substituting in Eq. (7),

$$\begin{aligned}\text{Db change} &= 50 \log_{10} {}^{660}\!/_{600} \\ &= 50 \times 0.0414 = 2.07\end{aligned}$$

Therefore the sound rating of this fan operating at the higher speed would be 40 + 2.07, or about 42 db.

Where it is desired to make a comparative measurement of the noise radiated from air-duct inlets and outlets, it has been considered good practice to locate the microphone of the acoustimeter 1 ft. away from the edge of the opening and at an angle of 45 deg. from the duct axis. In all cases the microphone must be kept out of the direct air stream.

The sound level does not necessarily indicate the degree of annoyance produced, because the annoyance is greater at higher frequencies. For example, the degree of annoyance is nearly equal for sound levels of 40, 55, and 68 db at frequencies of 5,000, 1,000, and 100 cycles, respectively.[2]

Figure 253 shows the approximate noise levels for various familiar conditions.

Exact limits for permissible noise are not as yet in common use in air-conditioning work. In general, it is regarded as acceptable if the equipment noise does not increase the noise level by more than 3 db in places that are already moderately noisy. From the logarithmic relationship previously explained, this means that the equipment noise alone, as it reaches the room, must not be greater than the room noise alone. If the equipment noise is

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[1] Sound Measurement Test Code for Centrifugal and Axial Fans, National Association of Fan Manufacturers, June, 1942.

[2] J. S. PARKINSON, "The Nature of Noise in Ventilating Systems and Methods for Its Elimination," *Jour. A.S.H. & V.E.*, March, 1937.

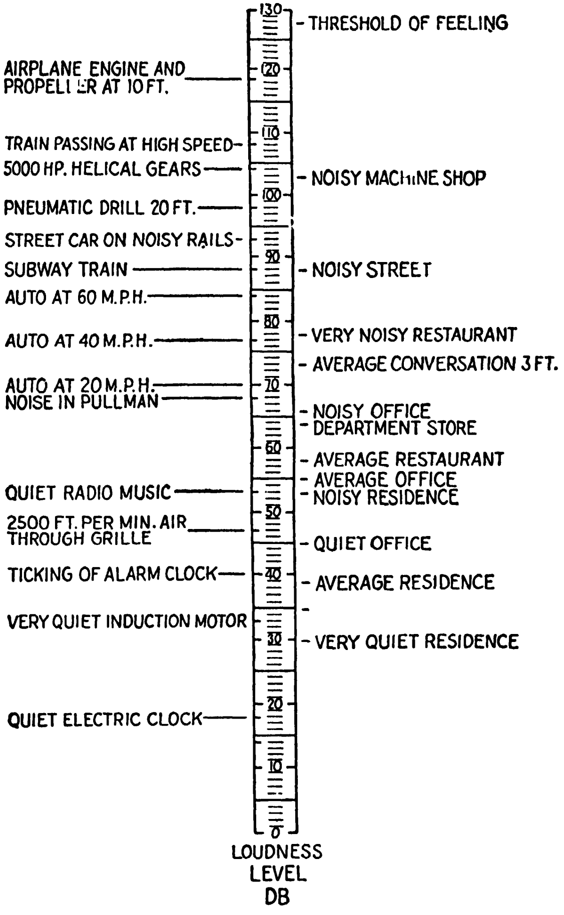

FIG. 253.—Loudness levels.

held to 10 db or more below the level of the room noise alone, the equipment noise will not usually be perceptible when added to the room noise.

312. Air Distribution. General.—The problem of introducing and distributing air in a room is without a doubt the most unsatisfactory phase of air conditioning at the present time. Many of the complaints about air-conditioned rooms are directly traceable to drafty distribution. The problem is not simple. The forces involved are extremely small, and the air currents are often erratic and unpredictable. Unless the air can be conveyed through ducts to several points of the room, it must be projected across the room by its own velocity; and if the ceiling is not very high, it is difficult to have these high velocities in the upper portion of the room and still avoid a higher rate of movement in the zone of occupancy than is permissible for comfort.

As to the permissible rate of air movement, a rate not exceeding 30 ft. per min. is necessary if ideal conditions are desired and where the occupants are seated. Somewhat higher rates of movement—up to 2 ft. per sec.—are permissible if the occupants are moving about. An air movement toward the faces of the occupants is much less uncomfortable than a movement from the rear. The *general* air movement is interrelated with the temperature and humidity in producing a particular effective temperature. *Localized* air movement on the ankles, neck, or head is usually uncomfortable regardless of the temperature or humidity.

In addition to freedom from drafts there is the requirement of reasonably uniform distribution of the air throughout the room in order to produce a uniform temperature and avoid the accumulation of odors.

313. Two Important Principles.—The first of these is the **injection effect** of a high-velocity air stream. When a stream of air is projected into a room in the manner shown in Fig. 254, it creates a secondary circulation as shown by the arrows *A*, *A*. This is due to the viscosity or internal friction of the air which causes the surrounding air to be dragged along by the moving stream. The circulation thus produced is often very pronounced. It is sometimes a distinct advantage in helping a stream of cold air to diffuse into and mix with the room air but in certain cases may be a disadvantage.

The second important phenomenon is the **decay** of the projected stream. The same property—viscosity—causes layers of the stream to be torn away, so to speak, in the manner of Fig. 254, as shown by the arrows *B*,*B*. The distance required for substantially complete decay of the air stream depends upon the initial velocity and upon the initial size and shape of the jet. In a large high-ceilinged room the supply grilles may thus be few and large, which would not be permissible in a low-ceilinged room. Although in a theater, for example, a few large grilles in the ceiling project cold air directly downward toward the audience, the long travel causes complete disintegration before the occupied zone is reached.

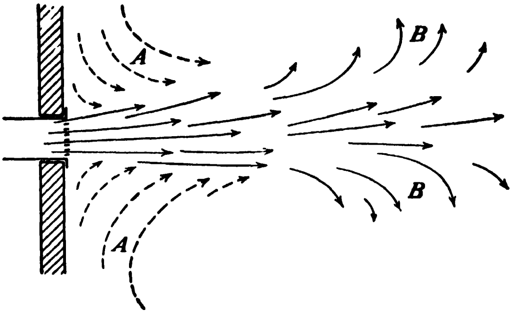

FIG. 254.—Secondary air circulation and the decay of an air stream.

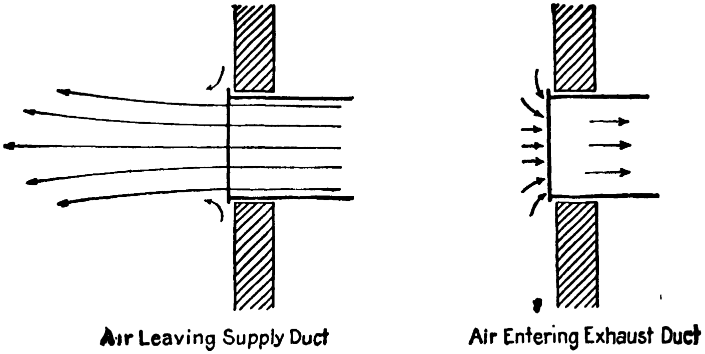

FIG. 255.

Although it is easy to project a stream of air several feet into a room from an *inlet* duct if a high velocity is used, the influence of an exhaust *outlet* extends for only a short distance. This is illustrated in Fig. 255 and is a point that should not be lost sight of. The flow of air toward an exhaust outlet is perceptible only in close proximity to the grille. Air can be projected into a space but can be withdrawn only from the vicinity of the outlet.

314. Approach Conditions.—Too little attention is often given to the conditions of approach to the supply grilles.

Regardless of the arrows on the engineer's drawing, the direction of the incoming air will not necessarily be perpendicular to the face of the grille if there is a turn in the duct. Figure 256*a* illustrates this point. When a supply duct is enclosed in a column or wall, it is sometimes constructed as shown, and the air, instead of entering horizontally as intended, is projected downward. The remedy shown in Fig. 256*b* is to use a longer sweep in the duct and to provide internal vanes.

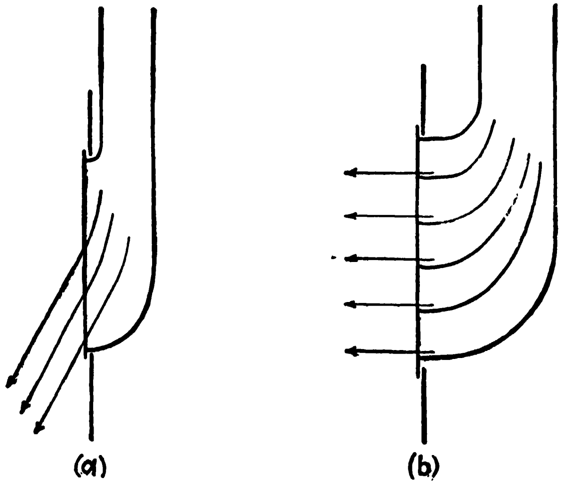

Fig. 256.—Incorrect and correct designs of inlets.

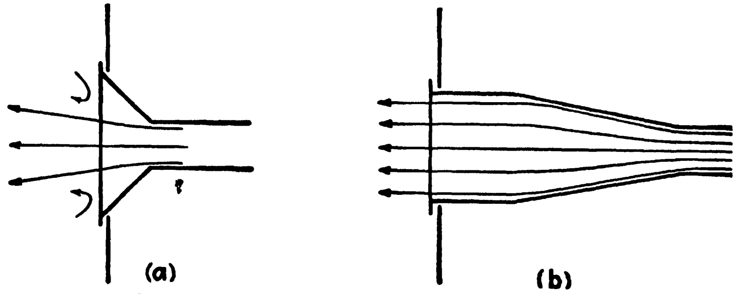

Fig. 257.—Effect of abrupt enlargement of duct.

When a low inlet velocity is desired, it is not sufficient merely to enlarge the outlet as in Fig. 257*a*, for the air will still be projected at high velocity from the central portion. The duct must be gradually enlarged some distance back of the grille. The natural angle of divergence of an air stream in the open is about 15 deg.

The *quantity* of air discharged from a grille depends upon the velocity and the effective area of the grille. The *velocity* depends upon the static pressure in the duct and upon the degree to which the static pressure is converted into velocity. It is often necessary to adjust separately the quantity of air and its velocity. The quantity is usually adjusted by a volume damper in the duct, and the velocity by properly choosing the size of grille.

Because of the lack of exact knowledge about the introduction of air into rooms it is sometimes desirable to provide for adjustment of the inlet velocities by providing grilles of ample size. The velocities can then be increased when the installation is finally tested, if necessary, by blocking off part of the grille opening. This is not common practice and naturally increases the cost of the system slightly, but it does provide a means of adjusting the *velocity* independently of the *volume.*

315. Methods of Air Distribution.—There are several methods of introducing air which are in common use and which can be reasonably successful if properly designed. The type of circulation used is often not so important as the care with which it is applied. The most common method for many kinds of rooms is that shown in Fig. 258, in which the supply and exhaust grilles are

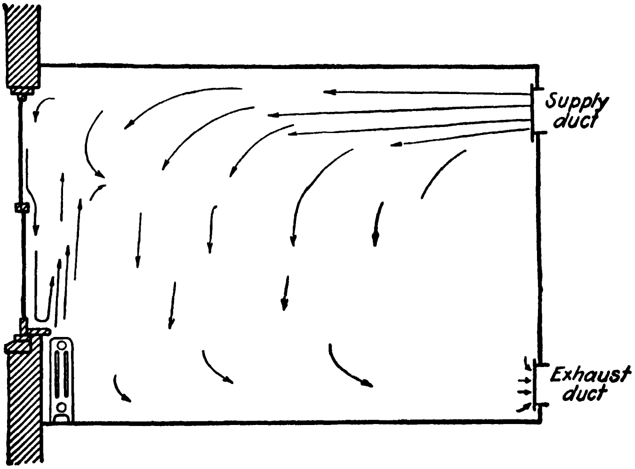

FIG. 258.—A common method of air introduction.

in the same wall and usually opposite the windows. The cut shows the radiator in action, but the same method is applicable for summer cooling systems. It is important that the velocity of the air leaving the grille be sufficient to carry across the room but not so great as to strike the opposite wall and cause a down-draft. Lateral spread of the air is obtained by vertical deflectors in the grille.

The direction of air movement will vary greatly depending upon the relation of the temperature of the incoming air to that of the room. If the incoming air is warm, it will tend to hug the ceiling and be displaced by the air following it. In summer, when the incoming air may be as much as 20° below the room air, it will tend to fall a short distance from the face of the grille unless the velocity is high.

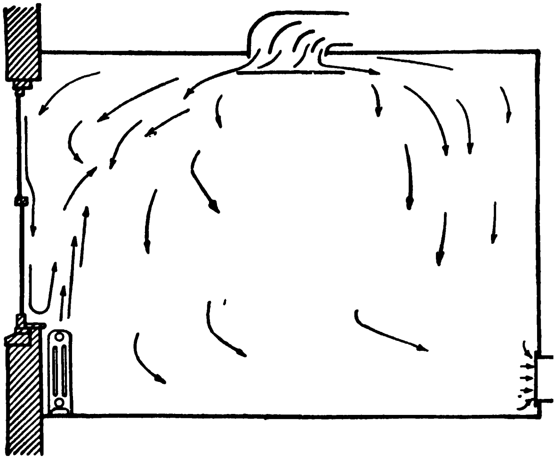

Fig. 259.—Ceiling inlet—pan system.

The velocity of the jet must therefore be sufficient to carry it out into the room but not so high as to cause it to impinge on the opposite wall and thus cause an objectionable downward and reverse current. In a long room it is sometimes desirable to have high-velocity grilles to project air to the far half of the room and low-velocity grilles to serve the near half. The velocity depends, of course, upon the static pressure behind the grille, which may be modified by properly designed dampers.

Side-wall introduction is probably used more than any other method, because it is simple and cheap. The results are reasonably satisfactory but leave room for improvement when air below room temperature is introduced.

A second scheme is shown in Fig. 259, in which the air enters the room near the center of the ceiling (or, in a large room, at spaced points) and strikes a plate that deflects it horizontally in all directions. This is a useful method when the ducts cannot be run in partitions or columns. The plate may be ornamented to harmonize with the decorations. It is important in this scheme to have a perfect downward approach, or the air will not distribute uniformly.

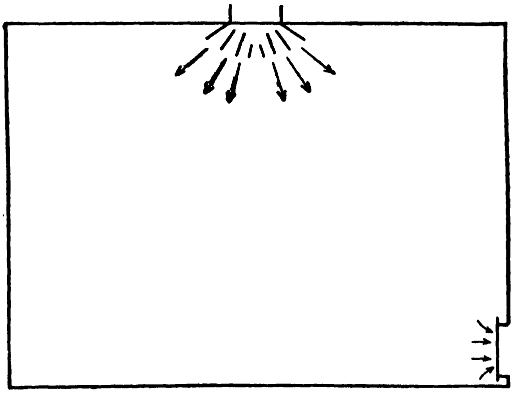

FIG. 260.—Rosette inlet.

Figure 260 illustrates a method in which the air is introduced from a rosette of concentric cones that serves to distribute it at a moderate velocity. Each supply point serves a square or nearly square portion of the room. Here also the approach conditions must be correct for the best operation.

Figure 261 shows a method not in common use but successful where large quantities of air must be introduced at temperatures below that of the room.[1] A section of the ceiling amounting to one-third or more of the ceiling area is constructed of thin, perforated plates through which the air, because of the large area, enters the room at an extremely low velocity. The supply ducts *A*, *A* extend across the entire room (perpendicular to the plane of the sketch), and the air passes through adjustable openings, spaced at intervals, into the diffusion chambers *B,B* above the perforated plates. The perforated plates may form a part of the acoustic treatment often used in offices. This method of introduction is not universally applicable.

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[1] J. N. LIVERMORE, "Unusual Air Conditioning Problems Met in Designing New Glass Block Building," *Heating, Piping and Air Conditioning*, April, 1938.

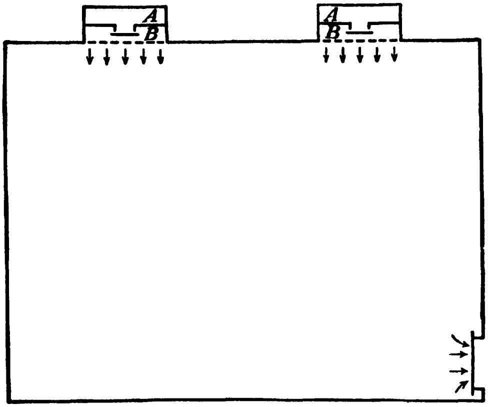

FIG. 261.—Porous-ceiling method.

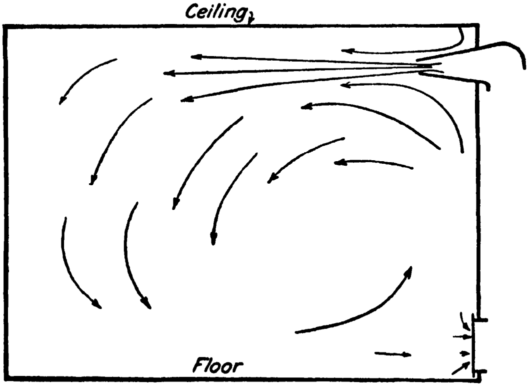

FIG. 262.—High-velocity injection.

Figure 262 shows an injector system in which a row of nozzles introduces air at a high velocity in order to create a large secondary circulation. This method as illustrated is used chiefly for industrial air conditioning, but the principle of injection is employed in certain special grilles used in comfort air conditioning.

Where ventilation only is desired, the inlets and outlets may be located as in Fig. 263. The incoming air must have a low velocity to avoid uncomfortable drafts. This method is used in many dining rooms, smoking rooms, etc. It is not suitable for a cooling system.

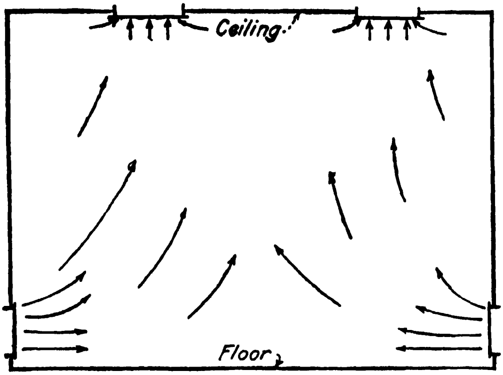

FIG. 263.—Upward ventilation.

316. Grilles.[1]—For supply grilles the ordinary grating, which merely serves the purpose of partially masking the opening, is suitable for simple ventilating systems but not for high-grade air-conditioning work, especially when cooling is to be done.

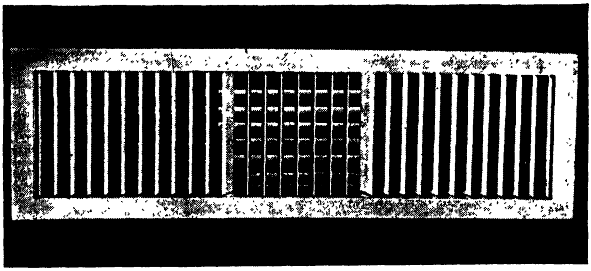

FIG. 264.—Grille with directional blades. (*Waterloo Register Company.*)

For that purpose some form of grille that controls the direction of the air is generally necessary to secure proper distribution. Figure 264 shows a grille that has vertical blades to adjust the horizontal spread of the jet and horizontal blades to adjust the angle in the vertical plane. Such a grille is desirable because it provides considerable flexibility, although in many cases grilles are used that have only vertical or only horizontal blades. In some designs the blades are fixed to give a predetermined directional effect.

---

[1] The term "grille" usually designates a barred or ornamented face for an inlet or outlet. A "register" is a grille combined with a damper for cutting off the air flow.

The velocity required for a desired distance of throw from the face of the grille depends upon the quantity of air, its temperature with relation to room temperature, and the design of the grille. The grille manufacturers' tables should be consulted. The relation between the velocity and the velocity and the static pressure behind the grille is approximately expressed by Eq. (2), Chap. XVI.

$$h_v = \left(\frac{V}{1{,}096.5}\right)^2 \times D$$

where h_v = velocity head (which is approximately equal to the static head behind the grille), in. of water.
V = velocity, ft. per min.
D = density of air, lb. per cu. ft.

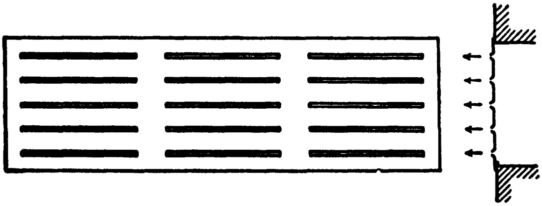

FIG. 265.—Injection-type grille.

A special type of grille, illustrated in Fig. 265, takes advantage of the injection principle previously discussed. The face of the grille consists of slots arranged either horizontally, as shown, or vertically and varying in width from $\frac{1}{8}$ to $\frac{1}{2}$ in. The edges are slightly curved to a nozzle shape. The air leaves at high velocity (up to 2,500 ft. per min.), and the injection effect causes a pronounced secondary circulation (Figs. 254 and 262) that serves to mix the incoming air with the room air. The incoming air may therefore be as much as 20 to 25° below the room air without causing serious discomfort.

The supply fitting shown in Fig. 266 is representative of a new type which is coming into use and which is applied in the manner shown in Fig. 260.

317. Static Pressures.—The velocity of the air leaving the grille and consequently the air quantity (with given grille area) depend upon the static pressures in the duct and immediately behind the grille. Therefore if the delivery to the rooms is to be maintained at the design figure, a reasonable constant static pressure must be carried in the supply duct. With a fixed fan speed and no changes in any damper settings the static pressures will naturally remain nearly constant; but this condition sometimes does not exist. For example, if the occupants in some of the rooms can close and open their supply dampers, the static will change in the supply duct, and the delivery to the other rooms will be affected, perhaps seriously enough to cause discomfort. The remedy is to use static pressure control.

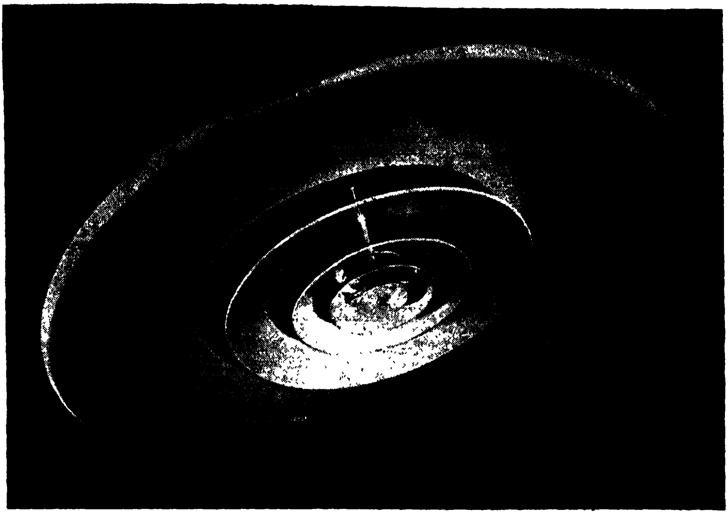

FIG. 266.—Rosette-type air supply fitting. (*Anemostat Corporation of America.*)

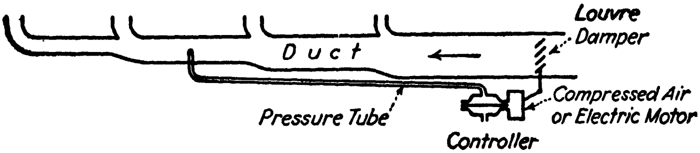

FIG. 267.—Static pressure control applied to a distribution duct.

Figure 267 shows diagrammatically a simple way of applying such a control. A controller, having a primary element consisting of a very sensitive diaphragm or float chamber, operates a louver damper in the duct by using compressed air or electricity as motive power, and the varying resistance of the damper serves to maintain a nearly constant static at some mid-point in the duct. Strictly speaking, it is the static difference between the duct and the rooms that is controlled. Such a control is not generally used but is desirable if ideal conditions are sought.

318. Removal of Air from Rooms.—The planning of the exhaust system and the location of exhaust grilles in the rooms are more important than is sometimes thought. The comfort conditions may be seriously affected by drafts caused by the exhaust system.

It is often economical to use corridors as exhaust ducts, and this is particularly true in office buildings. The space between false ceilings and the concrete floor above can also be used to advantage in some cases. The plastering must be tight, however, or else dirt streaks will appear near minute openings. Where any space other than a metal duct is used, it is necessary to avoid any accidental openings that might destroy the slight suction carried on that side of the system.

The chief point about the location of exhaust grilles is to arrange them so that none of the occupants will be stationed so close as to be in an uncomfortable draft. The air approaching the grille often has an appreciable velocity, particularly if the grille is a large one.

Velocities through exhaust grilles should be 500 to 750 ft. per min. The grille itself is usually a simple grid, intended merely to mask the opening. It is often desirable to supplement it with a fine-mesh screen to prevent accumulation of combustible rubbish in the ducts.

319. Measurement of Air Flow.—In ducts the flow of air is most accurately measured with the pitot tube. Several instruments are available for measuring flow at grilles and in the open.

The **anemometer** (Fig. 268) is a convenient instrument for measuring the flow of air at registers or where the velocities are too low for the ordinary pitot tube to give a measurable gage deflection. It is not suitable for extremely low velocities, as the power required to revolve the propeller then becomes the source of a considerable error. The maximum velocity for which it is suitable is about 600 ft. per min.

The anemometer is a velocity-measuring device, the dial being calibrated to read directly in feet. The velocity is computed by dividing the dial registration by the elapsed time between the readings.

In using the anemometer to measure the flow through an air-inlet grille or an exhaust grille, the face of the grille is marked off into a number of equal squares or rectangles, and an average taken of the readings for the several spaces. The question naturally arises as to whether the gross area of the grille or the net area should be used for the calculation of the air quantity. The following methods of computation give quite accurate results:[1]

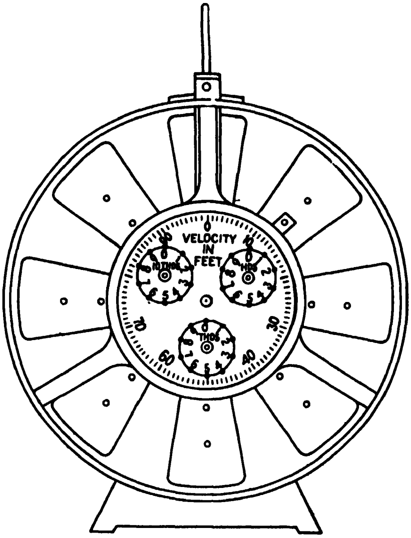

Fig. 268.—Anemometer.

For supply registers: Hold anemometer in fixed position in front of each division and 3 in. away from face of register. Compute air flow from the following formula, using the proper value of C from Table 86:

$$\text{C.f.m.} = \frac{CV(A + a)}{2} \tag{8}$$

---

[1] Lynn E. Davies, "The Measurement of the Flow of Air through Registers and Grilles," *Trans. A.S.H. & V.E.*, 1930.

where C = a constant, from Table 86.
V = velocity, measured by the anemometer.
A = gross area of grille.
a = net area of grille.

For exhaust registers: Hold anemometer in contact with the face of the grille and proceed otherwise as above.

TABLE 86.—VALUES OF C

Average indicated velocity, feet per minute	Supply grilles, C	Exhaust grilles, C
150	0.952	0.993
200	0.957	1.005
300	0.967	1.028
400	0.977	1.049
500	0.985	1.067
600	0.992	1.078
700	0.998	1.084
800	1.000	
Factor for average use:		
Low velocity	0.97	1.020
High velocity	1.00	

The following formula may be used for exhaust registers only:

$$\text{C.f.m.} = kva \tag{9}$$

Davies suggests values for k ranging from 0.762 at low velocities to 0.832 at high velocities. More recent work[1] with grilles of modern design indicates that a value for k of 0.873 is reasonably accurate for a considerable range of velocities and grille designs.

The **vane anemometer** (Fig. 269), a new instrument, is valuable because it gives an instantaneous reading. It consists of a delicately balanced and magnetically stabilized vane within a case through which the air flows when it is pointed into the air stream. It reads down to low velocities and is suitable for measuring in rooms air currents of about 50 ft. per min. Attachments may be used to adapt it to the measurement of flow in ducts.

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[1] A. M. GREENE, JR., and M.H. DEAN, "The Flow of Air through Exhaust Grilles," *Jour. A.S.H. & V.E.*, 1938.

The **hot-wire anemometer** is another new instrument, not yet in common use. It consists of a resistance wire heated electrically and in contact with a thermocouple or thermometer. A constant current is maintained through the wire to give a fixed heat output, and the observed temperature of the wire is an indication of the air movement when referred to the proper calibration data. A recently designed instrument[1] of this type is practically free from directional effects and, by the proper selection of heating currents furnished to the thermocouple, air velocities of 20 to 2,000 ft. per min. may be measured accurately.

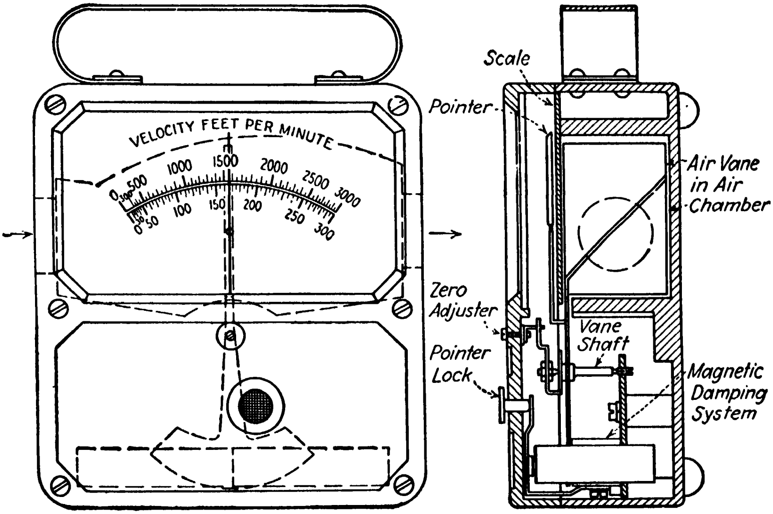

FIG. 269.—Vane anemometer. (*Illinois Testing Laboratories, Inc.*)

The dry **katathermometer** is a useful nondirectional device for measuring air movement in a room. It is essentially an alcohol thermometer with an oversize bulb. When a reading is taken, the bulb is warmed in water to above 100° and dried. The time required for the liquid level to fall from the 100° to the 95° mark is taken with a stop watch and referred to a calibration curve from which the velocity is read. (The same instrument with a wetted jacket around the bulb may be used to determine the cooling power of the atmosphere.)

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[1] KRATZ, HERSHEY, and ENGDAHL, "Development of Instruments for the Study of Air Distribution in Rooms," *Trans. A.S.H. & V.E.*, 1940.

Smoke is useful in observing the air currents in a room. Puffs of tobacco smoke or chemical smoke indicate local air currents, and smoke bombs ignited in the supply duct serve to show the over-all circulation of the air.

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Problems

1. Show how the equation $D = 1.265 \sqrt[5]{\frac{AB^3}{A + B}}$ (bottom of Table 81) is derived. It is based on the principle that the round duct and the rectangular duct are equivalent when both carry the same volume of air at the same pressure drop. Refer to Eq. (1) (Chap. IX), which is approximately, though not quite, true for air flow.

2. Compute the pipe sizes for a trunk-duct system similar to that in Fig. 250, except that the air quantities in the different sections on a 70° basis are as follows:

Section	Air Quantity C.F.M.
AB	19,000
BC	7,500
CD	2,000
BE	6,000
EF	4,000

Maximum air temperature 130°.

3. In Fig. 250 calculate the size of sections *BE* and *EF* for the conditions stated in Art. 307 and using the method described. Assume static head at *F* of 0.25 in.

4. In Fig. 251 calculate the duct illustrated for a temperature of 130° and an air quantity of 2,100 c.f.m. (at 130°). Use a duct of square cross section. Calculate the static pressure required at the entrance to the duct at the plenum chamber.

5. A 10- by 16-in. duct, 50 ft. long, delivers 1,800 c.f.m. at a temperature of 120° at the outlet. The duct is uninsulated and passes through a space that has a temperature of 50°.

a. What is the initial air temperature?

b. What is the initial air temperature if the duct is insulated outside with ½ in. of material for which $k = 0.38$?

6. Show that if two noise sources of equal levels are combined, the new noise level is 3 db higher than that of each noise taken separately.

7. A round duct 50 ft. long carries 4,800 c.f.m. at 70°. The initial static head is 0.25 in., and the static at the far end is 0.15. At the far end are two grilles each delivering 2,400 c.f.m. (*a*) If one grille is now completely closed and the initial static (0.25) is unchanged, what will be the flow through the grille which remains open? (*b*) If one grille originally handles two-thirds of the air and the other one-third, and the larger is closed, what will be the new flow through the smaller?

Life is Long it will
never came again.
oh lord, Give me Strength
I have got lost in your
world?
Lord either kill me
or give strength

CHAPTER XVIII

COOLING AND DEHUMIDIFYING APPARATUS

320. Present and Future Fields.—The cooling and dehumidifying of buildings is a form of air conditioning that has recently become of considerable importance, and it is probable that the future will see a steadily increasing demand for artificial cooling not only for industrial purposes but in office buildings, stores, theaters, and homes.

For office buildings the advantages of artificial cooling in summer are beginning to be realized. Increased working efficiency, greater cleanliness, elimination of a great deal of outside noise because windows are kept closed, and a certain advertising value are among the chief points in its favor.

In retail stores it has real business-getting value, especially for basement and first-floor salesrooms. In the home, although the expense is high, there is every prospect of its growing use in the future.

Theaters have found cooling systems to be absolutely essential if any sort of attendance is to be maintained during the summer.

The cooling of railway cars is proving to be of definite commercial advantage and is progressing rapidly.

The objection to cooling systems is naturally their rather high cost of both installation and operation, but there is some prospect of this being partially remedied with the development of new apparatus and methods.

In industrial process work, cooling and—what is usually more important—dehumidification are essential in some industries. The true purpose of air conditioning or cooling in any particular case should be thoroughly understood and the desired conditions well known before an attempt is made to design the system. Certain conditions are most desirable for comfort, and different conditions are required for different manufacturing processes. It is the purpose of this chapter to discuss the apparatus used to produce the cooling and dehumidifying effect.

321. Artificial Methods.—The methods of artificial comfort cooling fall into two classes, those that produce *primarily cooling* and those whose function is *primarily dehumidification.* This distinction is not rigid, because all methods produce both cooling and dehumidification to some extent. Comfort cooling methods may be classified as follows:

Cooling Methods:

- Evaporative cooling.
- Well water.
- Ice.
- Mechanical refrigeration.

Dehumidifying Methods:

- Dehydration by adsorption or absorption.

The apparatus installed to utilize well water, ice, and mechanical refrigeration are very similar in many respects. The chief difference is in the source of the cooling effect.

322. Evaporative Cooling.—It was indicated in Chap. XVI that cooling can be accomplished by evaporation, using the air washer, and that the reduction of temperature of the air flowing through the washer is a function of the initial wet-bulb depression and of the humidifying efficiency of the washer.

The field of evaporative cooling is limited because of the relatively small reduction of dry-bulb temperature that it can produce with the climatic conditions that prevail in many parts of this country and because it does no dehumidifying but, on the contrary, adds moisture to the air. Therefore it is not suitable where a large amount of sensible heat is to be removed from the cooled space or where the wet-bulb depression is small. This can be readily illustrated by examples.

Example 1.—(Evaporative cooling not appropriate.) Assume a room having lights, machinery, and heat gain through the structure amounting to the high figure of 350,000 B.t.u. per hr., the room having a volume of 100,000 cu. ft. Assume the outside air to have a temperature of 90° dry bulb and 75° wet bulb. It is desired to maintain an inside temperature of 80°.

The washer has a humidifying efficiency of 100 per cent, and the air will therefore be introduced at 75° saturated.

Carrier total heat of entering air 75° saturated = 37.7 B.t.u. per lb.

Carrier total heat of leaving air 80°, 85 per cent relative humidity[1] = 39.0 B.t.u. per lb.

---

[1] Assuming no moisture added in the room.

Difference = heat pickup per lb. = 1.3 B.t.u. per lb.

$$\frac{350{,}000}{1.3} = 269{,}000 \text{ lb. air per hr. required}$$

Specific vol. of sat. air at 75° = 13.88 cu. ft. per lb.

$$\frac{269{,}000 \times 13.88}{60 \text{ (min.)}} = 62{,}300 \text{ c.f.m. required}$$

$$\frac{62{,}300 \times 60 \text{ (min.)}}{100{,}000 \text{ (cu. ft.)}} = 37.3 \text{ air renewals per hour}$$

This is an unwieldy quantity of air to handle. It would require large fans and ducts and would be difficult to introduce without uncomfortable drafts. Note that the room conditions are 80° and 85 per cent relative humidity, giving an effective temperature of 78°, which is outside the comfort zone.

Example 2.—Now consider the same room but with a heat load of only 200,000 B.t.u. and with outdoor conditions of 90° dry bulb and 65° wet bulb. These conditions are not unreasonable for many localities in the Western states. The air would be introduced at 65° saturated.

Carrier total heat of entering air 65° sat. = 29.6 B.t.u. per lb.

Carrier total heat of leaving air 80°, 60 per cent relative humidity = 33.2 B.t.u. per lb.

Difference = heat pickup per lb. = 3.6 B.t.u. per lb.

$$\frac{200{,}000}{3.6} = 55{,}500 \text{ lb. air per hr. required.}$$

Specific vol. of saturated air at 65° = 13.5 cu. ft. per lb.

$$\frac{55{,}500 \times 13.5}{60 \text{ (min).}} = 12{,}500 \text{ c.f.m.}$$

$$\frac{12{,}500 \times 60 \text{ (min.)}}{100{,}000 \text{ (cu. ft.)}} = 7.5 \text{ air renewals per hr.}$$

The room conditions would be 80° dry bulb and 60 per cent relative humidity, which is just at the border of the summer comfort zone. The quantity of air is quite reasonable to handle.

Thus there is a definite though quite limited field for evaporative cooling. The conditions under which it is acceptable are (1) large wet-bulb depression; (2) moderate heat load per cubic foot of room volume; (3) low moisture load in the rooms; and (4), unless all other conditions are favorable, rather high effective temperature in the rooms as compared with ideal comfort.

It is therefore possible to use evaporative cooling during moderate weather and mechanical refrigeration during the period of extreme conditions of high outdoor wet bulb. This is rarely done but is perfectly feasible if the system is properly designed and if the operation is intelligently controlled.

323. Well Water.—Where deep-well water or any other source of water at a temperature of 60° or less is available, it is an ideal medium for cooling and dehumidifying, because it is usually the cheapest and is simple in its application. In the use of well water the air is simply passed over coils through which the water flows. In some localities it is reasonably certain that an ample flow of water can be had by driving a moderately deep well. In many places, however, well drilling is rather a risky enterprise, and its probable cost and chance of failure to find water must be considered. Also, in some areas the well water may be corrosive or contain objectionable gases.

Water at a temperature of 60° or less is necessary chiefly because of dehumidification requirements. With higher water temperatures the insufficient dehumidifying effect results in a high relative humidity that is not pleasant.[1]

324. Ice.—Cooling by means of ice has certain advantages. It requires a somewhat smaller investment in equipment than any method except the use of well water, and the apparatus is simple and free from operating troubles. Ice is particularly well adapted to the cooling of buildings that are infrequently occupied and where a relatively high hourly operating cost is less important than the low investment cost. Churches, for example, come in this class.

One objection to the use of ice is the trouble of procuring a quantity of it when cooling is required. With fluctuating summer weather it is not economical to maintain a charged ice tank throughout the summer, and the assurance of quick delivery on days when cooling is found to be needed is essential.

Cooling systems using ice are of two classes, the portable unit and the central-fan system. In the earlier days of air conditioning the portable unit appeared to have considerable promise, but it has not been generally accepted by the public. Portable units, designed to hold a charge of 300 to 500 lb. of ice, contain a fan that blows the air over the ice or over metal surfaces cooled by the ice. They are particularly suitable for occasional use, as in hospitals and sickrooms. Figure 270 illustrates a satisfactory arrangement for an ice tank. The ice is introduced through a sidewalk hatch directly into the tank located in the basement. The tank contains water in which the ice floats. A pump takes water partly from the tank and partly from the return water line and pumps it through the cooling coils located in the air circuit. By thus mixing the return water and cold water under thermostatic control, the coil temperature and consequently the dew point of the air supplied to the cooled rooms can be adjusted. The water returning to the tank is sprayed upward against the top of the tank and rains down over the surface of the ice cakes.

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[1] A thorough study of the subject is reported in "Summer Cooling in the Warm-air Heating Research Residence with Cold Water," *Univ. Ill. Eng. Exp. Sta. Bull.* 305.

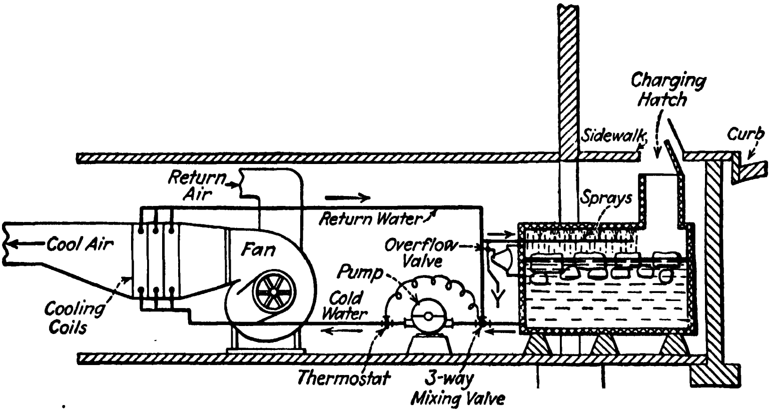

FIG. 270.—Cooling system using ice.

A float-controlled valve regulates the overflow so as to hold a constant level in the tank. The tank, pump, and piping should be well insulated to avoid condensation from the atmosphere and waste of ice.

An arrangement of this type, with the ice afloat in the tank, is better than placing the ice on a rack in the air, because it ensures complete wetting of the ice and uniform melting. There is no particular advantage in having a larger volume of water in the tank than just enough to float the ice. The tank should be large enough to hold somewhat more than the maximum amount of ice consumed between fillings, which should be made daily.

The temperature of the ice water will usually be between 35° and 40°, but the water to the coils can be carried at any temperature from the ice-water temperature to one approaching the return-air temperature, depending upon the cooling load.

325. Economics of Ice Cooling.—The latent heat of melting of ice is 144 B.t.u. per lb., and some additional heat is extracted from the ice water, so that the total heat available per pound is about 150 B.t.u., or 300,000 B.t.u. per ton. Ice usually sells, in quantities, at $4 to $6 per ton, making the cost per million B.t.u. of heat absorption from $13.33 to $20. A motor-driven compression machine would require about 117 kw.-hr. for 1,000,000 B.t.u. of refrigeration.[1] If electricity costs 2½ cents per kilowatt-hour, the cost per million B.t.u. for the compression machine would be $2.92 as compared with $13.33 to $20 for ice. There would be, however, the cost of cooling water for the compression machine, which would decrease this margin somewhat. The field for the use of ice is in occasional cooling or in reducing the first cost of the installation at the expense of a higher operating cost.

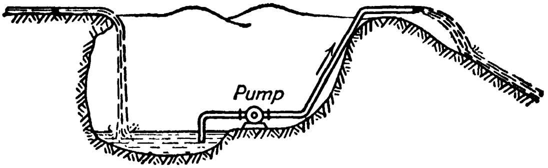

FIG. 271.—Mechanical analogy of a refrigerating machine.

326. Mechanical Refrigeration.—A refrigerating machine is simply a mechanical device for raising a given quantity of *heat* to a *higher temperature level* so that it can be carried away by some available medium such as the atmosphere or a stream of water. Thus a household electrical food refrigerator extracts heat from the food cabinet and discharges it to the surrounding atmosphere in the kitchen.

To use a simple mechanical analogy, consider the pump in Fig. 271, which is continually drawing water from a pool into which a stream is flowing and is pumping it over the crest of the adjacent hill. The water cannot flow out of the basin by gravity but must be lifted by the pump to the higher level before it can flow away. The pump corresponds to the refrigerating machine, and the basin corresponds to the cooled space into which heat is flowing from the outside. A refrigerating machine pumps heat uphill, so to speak.

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[1] Based on a coefficient of performance of 2.5 (see Art. 329 for definition).

In cooling a building the heat that is extracted could be discharged to the outside air; but it is usually more economical to carry it away by means of water, which is usually available at a lower temperature than the air. It requires more energy to elevate a given quantity of heat to a considerably higher temperature than to one only somewhat higher.

327. Compression Refrigerating Machine.—Mechanical refrigerating machines are of two types, the compression type and the absorption type. In air-conditioning work the former is used almost exclusively at present.

Figure 272 is a diagrammatic representation of the compression refrigeration cycle. The refrigerant, in a gaseous state, is compressed in the compressor and passes into the condenser, where it is cooled and condensed, and is then accumulated in the receiver. The liquid refrigerant flows to the expansion valve at the inlet of the evaporator. The expansion valve is simply a pressure-reducing valve that allows the refrigerant to pass through at such a rate as to maintain a predetermined low pressure in the evaporator coil. The reduction of pressure causes the refrigerant to evaporate, and the heat of vaporization is absorbed from the fluid (air or water) that is flowing over the evaporator coil. The temperature attained in the evaporator depends primarily upon the pressure-temperature characteristics of the particular refrigerant used. From the evaporator the now superheated gas flows to the compressor.

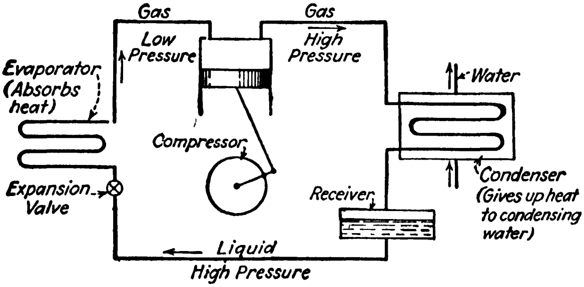

FIG. 272.—Compression refrigeration cycle.

It is evident that this process of refrigeration depends upon the evaporation and condensing of the refrigerant and upon the fact that the latent heat of such substances is relatively large.

Although it is possible to employ a gas, or mixture of gases such as air, which is not condensed and evaporated, the volume of gas to be handled would be extremely great, because only the sensible heat in the gas and not the latent heat would then be employed.

The *evaporator* is the heat-absorbing element of the system. In a large air-conditioning system the evaporator coil is surrounded by water, which after being cooled is circulated through the air washer or cooling coils placed in the air circuit of the system. In small systems, particularly in those using cabinets placed in the cooled space, the evaporator coil is placed directly in the air stream. This arrangement is known as the direct-expansion system. It is permissible only when the refrigerant used would be relatively harmless if it should accidentally escape into the air stream.

328. The Ton of Refrigeration.—The capacity of refrigerating machines and the rate of output are commonly expressed in the unit known as the ton of refrigeration. A ton of refrigeration is a rate of heat absorption equivalent to the latent heat in a ton of ice melted in 24 hr. Since the latent heat of ice is 144 B.t.u. per lb., 1 ton of refrigeration equals 144 × 2,000, or 288,000 B.t.u. per 24 hr. This is equal to 12,000 B.t.u. per hr., or 200 B.t.u. per min.

The ton of refrigeration was originally adopted as a convenient unit in ice-making practice. It is not particularly appropriate in air-conditioning work but is generally accepted and will probably continue to be used. Refrigerating machines are usually rated in tons of refrigeration based on standard conditions, namely, an evaporating temperature of 5°, a condensing temperature of 86°, a suction-gas temperature of 14°, and a liquid temperature of 77° before the expansion valve.

329. Coefficient of Performance.—The coefficient of performance is the ratio of the heat absorbed in the evaporator to the heat equivalent of the external work supplied by the compressor to the refrigerant. It does not take into account the imperfect mechanical efficiency of the compressor or of the engine or motor that drives it.

The *theoretical* horsepower per ton of refrigeration is calculated from the coefficient of performance as follows:

$$\text{Hp} = \frac{200 \times 778.26}{33{,}000 \times \text{cp.}} \tag{1}$$

where 200 = B.t.u. per min. for 1 ton of refrigeration.
778.26 = ft.-lb. equivalent to 1 B.t.u.
33,000 = ft.-lb. per min. in 1 hp.
cp. = coefficient of performance.

330. Refrigerants.—A large number of substances can be used as refrigerants, but only a few are especially suitable for any particular type of service. The criteria for a refrigerant are (1) its pressure-temperature relationships, (2) its latent-heat content, (3) its harmlessness to human beings if it should escape to the atmosphere, and (4) its cost. The first two of these points and their related factors will be understood by a study of Table 87, which shows the characteristics of the refrigerants that are commonly used in air-conditioning work. Ammonia and carbon dioxide are included merely for comparison.

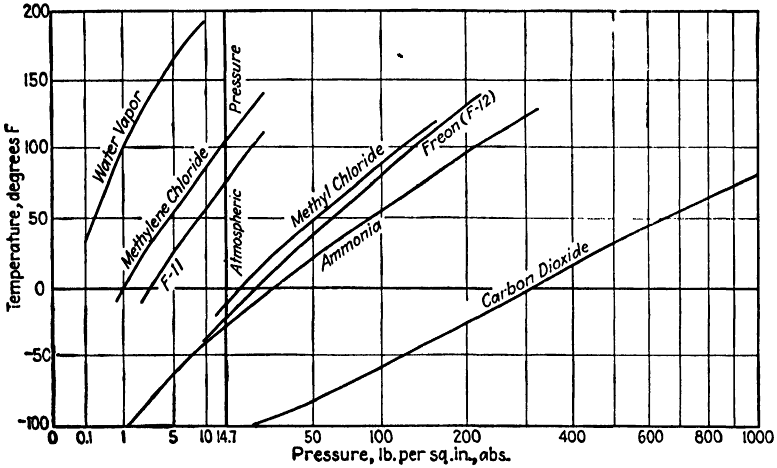

FIG. 273.—Pressure-temperature characteristics of refrigerants.

In this table the characteristics are given for the standard cycle of 5 to 86°, and some values are included for 35° and 100°, which are roughly the temperatures of the evaporator and condenser in a compression system designed for air conditioning. The chart in Fig. 273 shows the pressure-temperature characteristics (boiling points) of several refrigerants. It will be observed that the pressures required for water vapor are extremely low and that the pressures for carbon dioxide are very high, the others falling between. The pressure required for a refrig-

TABLE 87.—CHARACTERISTICS OF REFRIGERANTS

		Water vapor H_2O	Ammonia NH_3	Carbon dioxide CO_2	Methyl chloride CH_3Cl	Dichlorodifluoromethane, Freon, F-12 CCl_2F_2	Methylene chloride, Carrene No. 1 CH_2Cl_2	Monofluorotrichloromethane, Carrene No. 2; F-11 CCl_3F
Pressure of saturated vapor, lb. per sq. in. abs.	5°	....	34.27	331.95	21.0	26.51	1.17	2.98
	35°	0.1000*	66.26	529.3	38.7	47.27	2.96	6.30
	86°	0.6153	169.2	1,043.0	95.5	107.9	10.06	
	100°	0.9487	211.9	....	118.8	131.6	13.75	
Specific volume of saturated vapor, cu. ft. per lb.	5°	....	8.150	0.2660	4.65	1.485	49.9	12.1
	35°	2,947*	4.373	0.1576	2.54	0.863	21.1	6.08
	86°	527.7	1.772	0.0479	1.04	0.389	6.68	2.24
	100°	350.8	1.419	....	0.85	0.319	5.14	1.76
Heat of vaporization at 35°, B.t.u. per lb.		1,074.1*	540.5	98.8	173.65	66.28	155.59	81.2
Coefficient of performance† for standard cycle 5°–86° (for theoretical cycle = 5.74)†‡		4.10	4.85	2.56	4.67	4.72	5.14	4.82
Displacement per minute per ton of refrigeration, for standard cycle, cu. ft.		1,927§	3.43§	0.942§	6.83§	5.81	73.5	37.2
Theoretical horsepower per ton of refrigeration† (for theoretical cycle = 0.821)†‡		1.15	0.973	1.84	1.01	0.997	0.918	0.98

* From KEENAN and KEYES, "Thermodynamic Properties of Steam," published by John Wiley & Sons, Inc.

† Carrier and Waterfill.

‡ That is, for ideal Carnot cycle.

§ From W. H. MOTZ, "Principles of Refrigeration."

erant has a marked effect on the volume of gas that must be compressed and consequently on the size of the compressor. This is indicated by the displacement required in cubic feet per minute, as given in the table. Thus it will be readily understood that the refrigerant to be used has a decided influence on the design of the equipment.

The actual performance of refrigerating machines in horsepower per ton is somewhat different from the figures given in Table 87. The temperature cycle used in air conditioning calls for less power, but the imperfect mechanical efficiency of the compressor must be taken into account. For small units (2 to 5 tons) using Freon, the horsepower input per ton of refrigeration is about 1.2. For large machines using refrigerants such as methylene chloride, an average figure is 0.8 hp. per ton.

331. Comparison of Refrigerants.—Of the several refrigerants in Table 87, **water vapor** has the advantage of being cheap, perfectly harmless, and easy to handle. Its disadvantages are the difficulty of maintaining a perfectly tight system with the extremely low pressures that are necessary, and the relatively enormous volumes of vapor that must be handled. Because of the latter condition the reciprocating compressor is impracticable and the steam-jet or centrifugal compressor is used.

Ammonia (NH_3) is the standard refrigerant for ice making but is rarely used in air-conditioning work. Its suitability for ice making lies in its favorable pressure-temperature relationship. For air conditioning its extremely irritating and toxic effect on the human respiratory system is a serious danger in case of leakage.

Carbon dioxide (CO_2) is suitable for air-conditioning work and is sometimes used. The high pressures required, however, call for a rather expensive construction of the compressor and piping and necessitate careful operation and maintenance. Carbon dioxide is being superseded, in air conditioning, by refrigerants that do not require high pressures.

Methyl chloride (CH_3Cl) requires moderate pressures and was formerly used in air-conditioning work, particularly in small units, but is now seldom used. It is noncombustible at ordinary temperatures but can be ignited. It has a slightly toxic effect on the human system.

Dichlorodifluoromethane (CCl_2F_2), also known under the commercial name of Freon, or F-12, is widely and increasingly used in small units because of its suitable pressure range and its presumably harmless effect on human beings. It is used in many self-contained cooling cabinets and in large central systems.

Methylene chloride (CH_2Cl_2), also known commercially as Carrene No. 1, is particularly adapted for use in a centrifugal compressor because of the large volumes that must be handled. It is used particularly in large systems because the centrifugal compressor is expensive to build in small sizes. A capacity of 18 tons is about the lower economic limit.

Monofluorotrichloromethane (CCl_3F), also known as F-11 and Carrene No. 2, is coming into greater use because of its adaptability to a wide range of temperatures and its suitability for centrifugal compression. It is relatively nontoxic.

332. Toxicity of Refrigerants.—The hazard of refrigerants' escaping into the air of an occupied building has been given considerable attention by public authorities With regard to their toxicity, the National Board of Fire Underwriters classifies refrigerants as follows:[1]

Group 2.[2] **Ammonia.** Lethal or injurious in approximately ½ hr. in concentrations of 0.5 to 1 per cent.

Group 4. **Methyl chloride.** Lethal or injurious in approximately 2 hr. in concentrations of 2 to 2.5 per cent. (**Methylene chloride** reported as somewhat less toxic than Group 4.)

Group 5. **Monofluorotrichloromethane** (F-11) and **carbon dioxide.** Much less toxic than Group 4 and more so than Group 6.

Group 6. **Dichlorodifluoromethane** (F-12). Does not appear to produce injury in concentrations up to at least 20 per cent by volume in 2 hr.

Many refrigerants decompose into highly toxic compounds when allowed to come in contact with certain kinds of exposed flames.

333. Choice of a Refrigerant.—From a consideration of Fig. 273 and Table 87 it will be evident that the first consideration with regard to a refrigerant, from the standpoint of the design of the refrigeration system, is the pressure-temperature relationship at the boiling point. For the temperatures required, this relationship fixes the pressures for which the system must be designed and has an influence on the type of compressor used. The pressures required, together with the thermodynamic properties of the refrigerant, determine the volume that must be handled.

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[1] "The Comparative Life, Fire, and Explosion Hazards of Common Refrigerants," N.B.F.U., 1933. See also A.S.A. Safety Code for Mechanical Refrigeration, 1939 or later.

[2] Groups 1 and 3 in N.B.F.U. classification are not relevant to this discussion.

The present trend is toward the use of the low-pressure refrigerants in air conditioning, and carbon dioxide is passing out of use. Centrifugal compression, especially for the larger units, is gaining in favor. The steam-jet system is also favored when it is economically justified.

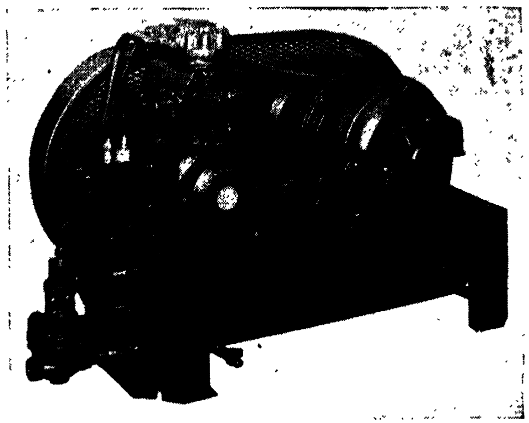

FIG. 274.—Condensing unit, water-cooled type with shell-and coil-condenser. (*General Electric Company.*)

334. Compressors.—For units of ½ ton to 100 tons or thereabouts, reciprocating compressors are available. Most of them now use F-12 as the refrigerant, are water cooled, and in the smaller sizes have the condenser assembled integrally with the compressor.

Figure 274 shows a 5-hp. unit, and Fig. 275 shows a four-cylinder unit of the larger type.

335. The Steam-jet System.—The particular suitability of water vapor as a refrigerant for air-conditioning work has led to the development and wide use of the steam-jet system. It is a compression cycle operating fundamentally like the other compression systems but using a steam-jet aspirator as the compressor.

Figure 276 is a diagram of one design of steam-jet system. The water returning from the coils of the air-conditioning system at a temperature of 55° or lower is sprayed into the cold tank, which is the evaporator. The pressure in the cold tank is reduced to a very low point (29.637 in. vacuum for 50° temperature) by the steam jet, and the water boils at this low temperature. The latent heat of vaporization is taken from the water itself, and the water temperature is reduced to 50° or less, depending upon the pressure. The chilled water is withdrawn from the

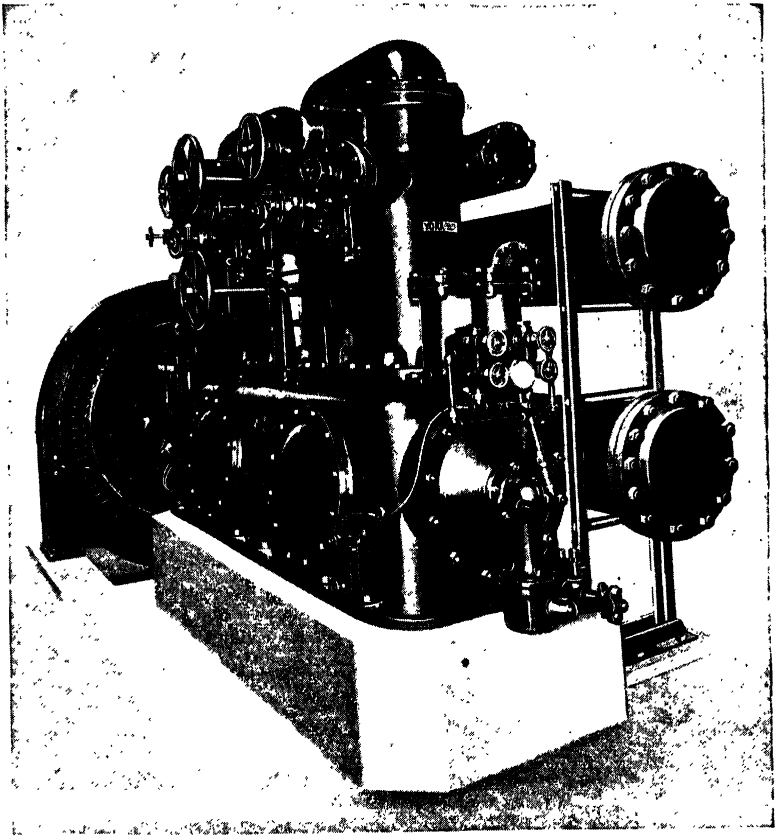

FIG. 275.—Large reciprocating condensing unit. (*York Corporation.*)

cold tank by a pump and circulated to the cooling coils in the air-conditioning system.

The steam-jet compressor consists of one or more (in this design, several) nozzles that discharge steam at a velocity of 3,500 to 4,500 ft. per sec. and, by an aspirating effect, draw vapor from the cold tank and force it through a carefully designed throat into a surface condenser in which a vacuum of about 28 in. is maintained. Condenser water flows through the tubes of the condenser, leaving at a temperature of approximately 98°.

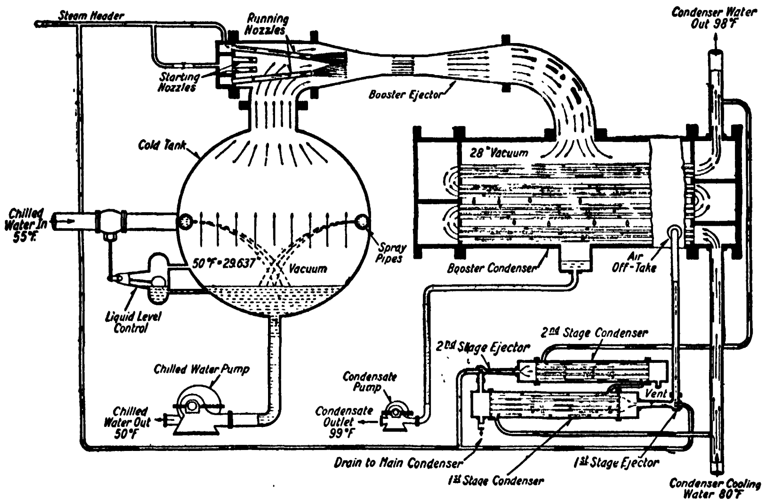

FIG. 276.—Diagram of steam-jet system. (*Westinghouse Manufacturing Corporation.*)

For economical operation at varying loads it is necessary to have several nozzles, because the working range of each nozzle is quite limited.

The small amount of air entrained with the steam must be removed from the condenser. The presence of this air is due to inleakage and to the air in the system on starting. It is drawn off usually in two stages, by auxiliary ejectors, that have their own condensers.

The purpose of maintaining a subatmospheric pressure in the main condenser is to decrease the compression ratio of the main ejector and thus permit it to operate at a reasonable steam consumption. On the other hand, the condenser pressure must be high enough to provide, for the temperature of the cooling water available, sufficient temperature differential between the steam and water to condense the steam, without requiring a condenser of excessive size.

The steam jet is simply one form of compressor. It is an economical method, from the standpoint of initial cost and operating cost, of compressing large volumes of gas. It is feasible to use it only when water vapor is the refrigerant, because it necessitates mixing the refrigerant with the steam. If an expensive refrigerant were used, it would be necessary to separate it from the steam after condensation.

The economic justification for choosing a steam-jet system in preference to a mechanical compression system will depend, in any given case, upon the relative costs of electricity, steam, and water. It is obvious that a much greater quantity of water is required in the steam-jet system, because it must condense not only the refrigerant but also the steam used in the jets. The amount of water used is three to four times that required in mechanical compression systems.

TABLE 88.—PROPERTIES OF STEAM AT LOW PRESSURES[1]

Temp. t, °F.	Abs. pressure p		Specific vol. of sat. vapor v_g, cu. ft. per lb.	Latent heat of vaporization h_{fg}, B.t.u. per lb.
	Lb. per sq. in.	In. Hg		
35	0.09995	0.2035	2,947.0	1074.1
40	0.12170	0.2478	2,444.0	1071.3
45	0.14752	0.3004	2,036.4	1068.4
50	0.17811	0.3626	1,703.2	1065.6
55	0.2141	0.4359	1,430.7	1062.7
60	0.2563	0.5218	1,206.7	1059.9
65	0.3056	0.6222	1,021.4	1057.1
70	0.3631	0.7392	867.9	1054.3
75	0.4298	0.8750	740.0	1051.5
80	0.5069	1.0321	633.1	1048.6

The simplicity of operation and maintenance and the relative costs usually form the basis of choice between the two methods.

The steam consumption of a steam-jet system depends upon the compression ratio (condenser pressure to evaporate pressure) and upon the initial steam pressure. These relationships are shown by the curves in Fig. 277. Thus for a condenser pressure of 3 in. Hg (27 in. vacuum) and a chilled water temperature of 50°, the steam consumption at 100 lb. gage would be 32.5 × 1.035 = 33.6 lb. per hr. per ton.

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[1] Reprinted, by permission, from "Thermodynamic Properties of Steam," by J. H. Keenan and F. G. Keyes, published by John Wiley & Sons, Inc., 1936.

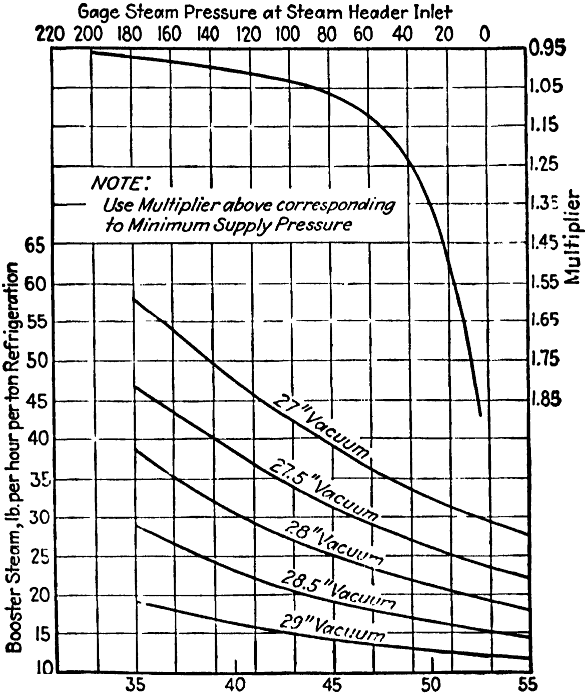

FIG. 277.—Steam consumption of a steam-jet refrigerating system. (*Westinghouse Manufacturing Corporation.*)

Table 88 is a portion of the steam table that shows the properties of steam at the very low pressures used in steam-jet systems.

336. Centrifugal Compression.—The centrifugal compressor is suitable for use with refrigerants that require the handling of large volumes of gas through a small compression ratio. At the present time the principal application of the centrifugal compressor in air conditioning is with the refrigerants methylene chloride, F-11, and water vapor. It is used in many air-conditioning installations both for comfort cooling and for industrial air conditioning. Figure 278 shows a commercial centrifugal unit for use with methylene chloride or F-11, in which the condenser and evaporator are incorporated, making the unit self-contained.

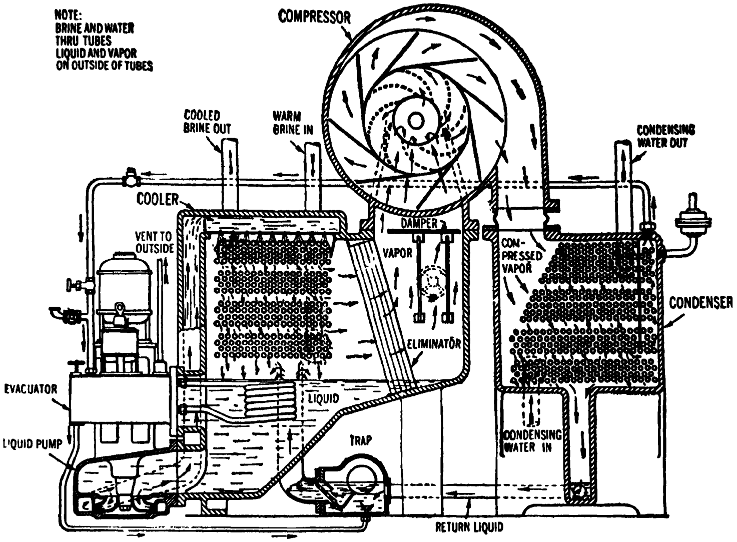

FIG. 278.—Centrifugal refrigerating unit. (*Carrier Corporation.*)

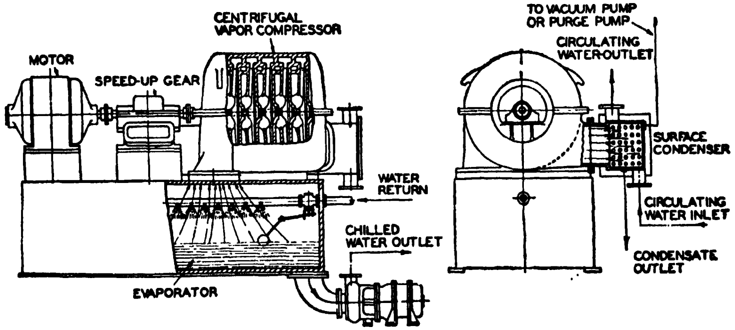

FIG. 279.—Centrifugal refrigerating unit using water vapor. (*Ingersoll Rand Company.*)

Because of its high speed the centrifugal compressor is suitable for turbine drive, and in some cases steam at pressures as low as 1 lb. gage has been successfully used to drive the turbine. Step-up gears are necessary when it is used with a motor drive. The compressor speeds are 3,000 to 5,000 r.p.m.

The centrifugal compressor with water vapor as the refrigerant has many of the advantages of the steam-jet system and the further advantage that there is only the refrigerant to be condensed and not the jet steam as in the steam-jet system. The centrifugal compression system using water vapor is of recent commercial development but may become widely used. Figure 279 shows a motor-driven unit.

337. The Absorption Refrigerating System.—The absorption refrigerating cycle, well known in commercial refrigerating practice, is gradually receiving some acceptance for air conditioning.

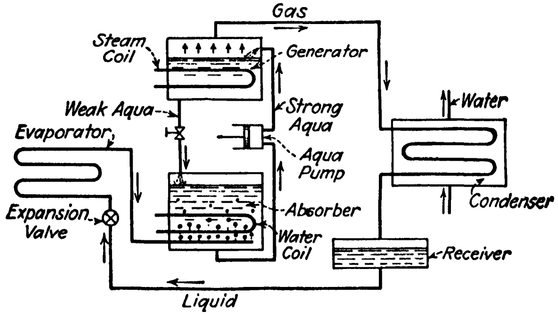

FIG. 280.—Absorption refrigeration cycle.

Basically the absorption system is somewhat similar to the compression system. It differs in that the compressor is replaced by a vessel for absorbing the refrigerant in water and another vessel for driving off the refrigerant in gaseous form from the water.

Figure 280 illustrates diagrammatically an elementary absorption system as used with ammonia as the refrigerant. The gas from the evaporator is brought into contact with the weak aqua (water containing little ammonia) in the absorber. The gas is absorbed, and the heat of absorption and the latent heat are removed by a water-cooling coil. The strong aqua, heavily laden with ammonia, is then pumped to the generator, which is a vessel containing a steam coil. The gas is boiled off by heat from the steam coil, and the weak aqua is returned through a regulating valve to the absorber. The gas passes from the generator to the condenser, where its latent heat is removed, and the condensed refrigerant flows from the receiver through the expansion valve to the evaporator.

The chief advantage of the absorption system for air conditioning is the fact that it uses only a small amount of power and instead employs heat as its principal form of energy. Therefore it has a certain economic advantage when heat is readily available. Although ammonia is not a desirable refrigerant, similar cycles using other refrigerants have been proposed.

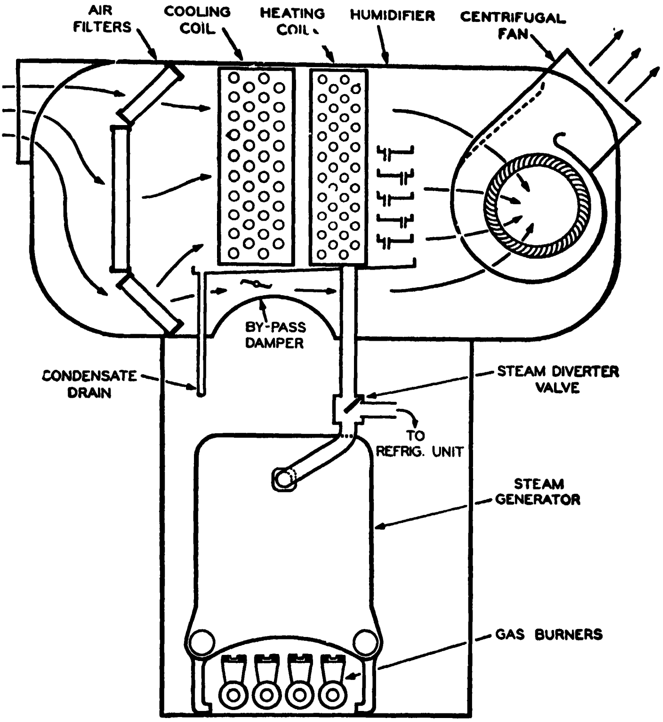

FIG. 281.—Sectional view of gas air-conditioning unit. (*Servel, Inc.*)

One of these cycles[1] utilizes lithium bromide as the absorbent and water as the refrigerant. The unit diagrammed in Fig. 281 is designed to operate under a vacuum and combines in a self-contained gas-operated unit all the functions of heating and cooling. This particular design has at present a capacity range of 3 to 5 tons of refrigeration.

The absorption cycle incorporated in this unit and schematically shown in Fig. 282 uses steam from an atmospheric-type cast-iron boiler for evaporating and lifting the solution in the generator to the separating chamber. A pressure of about 1.0 lb. per sq. in. is maintained in the generator and condenser elements of the cycle. The refrigerant (water vapor) passes to the water-cooled condenser and then flows through a U tube, which maintains the necessary pressure differential between the condenser and evaporator or cooling coil. Part of the water accumulating in the flash chamber vaporizes, thus tending to lower the refrigerant temperature before it enters the cooling coil. An absolute pressure of approximately 0.15 lb. per sq. in. is maintained in the evaporator which lowers the boiling temperature of the water to about 46°.

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[1] R. S. TAYLOR, "Heat Operated Absorption Units," *Refrig. Eng.*, March, 1945, and "Gas Summer Air Conditioning," *Am. Gas Assoc. Research. Bull.* 18, 1943.

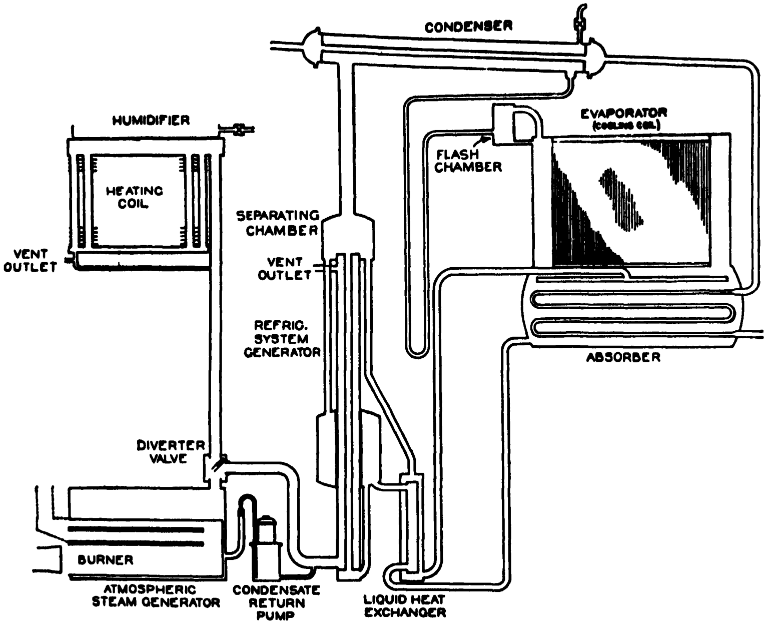

FIG. 282.—Diagrammatic arrangement of gas air-conditioning unit. (*Servel Inc.*)

The lithium bromide separated from the refrigerant by boiling action in the generator flows by gravity through a heat exchanger to the water-cooled absorber, which is located under the evaporator. Lithium bromide having a high affinity for water vapor absorbs it back into solution, and the resultant mixture drains back through the heat exchanger to the generator. The absorption process taking place creates a slightly lower pressure in the absorber than in the evaporator, and as a consequence the water vapor is readily drawn into the absorber. By taking advantage of differences in fluid temperature, density, and column heights, continuous flow through all circuits is accomplished without moving parts, except for the small pump needed to return the generator condensate to the boiler.

The function of the heat exchanger is to precool the lithium bromide before it enters the absorber, as its affinity for water vapor is greatly increased as the temperature is reduced. Similarly, the exchanger effectively preheats the cool absorbent-refrigerant mixture before it enters the generators, thus performing a dual function.

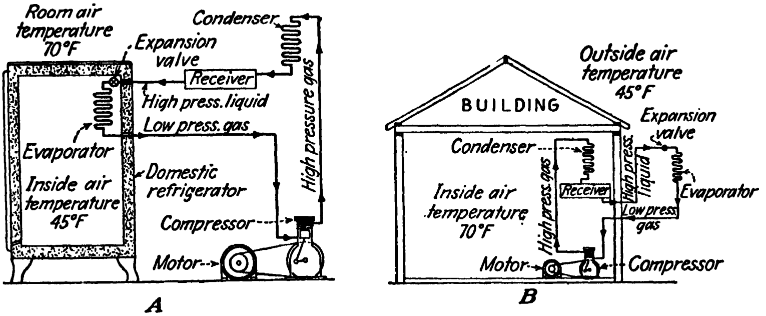

FIG. 283.—Reversed refrigeration cycle.

338. Reversing Cycle.—It is possible to use a refrigerating machine to accomplish winter heating. In such a system the refrigerant cycle is not actually reversed, but the heat, instead of being taken from indoor air and discharged to the air or water condenser, is taken from the outdoor air or well water and supplied to the enclosure through the release of heat from the condenser.

The diagrams shown in Fig. 283 illustrate the principles involved by comparing the operation of an ordinary domestic refrigerator with that of a reversed refrigeration cycle. The mechanical refrigerator is taking heat energy at a low temperature level of 45° and delivering it to a higher level of room air at 70°.

The reversed cycle arrangement shown in sketch *B* is illustrated by substituting a building for the refrigerator cabinet shown in diagram A and by exchanging positions of the evaporator or cooling coil and condenser or heat exchanger. Now the cooling coil is outdoors and the heat exchanger is inside the building. If the outside air is 45° and the desired inside air temperature is 70°, then by operating the compressor as a heat pump, just as in summer operation, heat energy will be picked up at a low temperature level and delivered at a higher temperature level.

The principal limitation to reversed cycle heating is the lack of reliable sources of heat. Outside air may be used; but when the most heat is needed, the outdoor air temperature is lowest, thus resulting in the least favorable combination. In localities where well water is available at a reasonably high temperature, it is more economical to use this source of heat. One system design[1] suggests using a combination of air and water, with the latter being used when the air temperature drops below 20°.

It is probable that the cost of supplying heat to a building by means of the reversed cycle would be economical only in rare cases. If the refrigerating machine is motor-driven, the power cost may be readily compared with the cost of supplying the same amount of heat by means of electricity used directly in resistance heaters. For such systems coefficients of performance from 6 to 8 are theoretically attainable, but under practical conditions a cycle of this design has a coefficient of performance as low as 3. This means that the heat supplied to a building would be three times the equivalent of the electric energy used to drive the motor. Therefore the energy used for heating by refrigeration would be at least one-third of that used for direct heating by electricity.

The field of the reversed cycle—if it may be said to have a field—is limited to the warmer climates where the size of refrigeration unit required for cooling is approximately equal to that needed for heating.

339. Storage Refrigeration.—In general, air-conditioning systems have a low annual load factor with a high peak-load demand, and as a consequence power rates per kilowatt-hour for this type of service are comparatively high. Methods of storing or accumulating refrigeration during off-peak periods have been devised that may benefit the customer as well as make air conditioning a more desirable load for electric utilities.

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[1] P. Sporn and E. R. Ambrose, "Description and Performance of Two Heat Pump Air Conditioning Systems," *Jour. A.S.H. & V.E.*, June, 1944.

Some of the methods thus far adopted include the storage of refrigeration in the form of chilled water, brine, or ice on evaporator plates. All of these require large tanks and for that reason have been found objectionable where space is limited. A unit-type cold accumulator, requiring less space, is arranged to form thin sheets of ice on metal plates in a steel tank.

With any of the above methods a smaller compressor may be utilized and operated for longer periods to produce the required refrigeration load. Storage refrigeration has certain advantages where the load is extremely variable, as in hotels, restaurants, churches, and mortuaries, but a careful economic study is necessary to determine whether such a design is practicable.

340. Dehumidifying Methods.—In Art. 321 it was explained that well water, ice, and mechanical refrigeration produce primarily a *cooling effect* and that other methods may be used whose effect is primarily *dehumidification.* This distinction is not rigid, because the former systems dehumidify by condensing moisture from the air, and the latter, as will be shown, can be used to cool the air to some extent. In either case the object in comfort work is to lower the effective temperature, which the one method does by attaining a lower dry-bulb temperature with a higher humidity, the other by means of a higher temperature and a lower humidity.

Dehumidifying methods, as now classified, dehydrate the air by **absorbing** or **adsorbing** the water vapor by means of substances having a strong affinity for water.

Absorption methods use water solutions of highly soluble chemicals. The solution tries, so to speak, to become more dilute; and when air is brought into contact with it, moisture is absorbed from the air by the solution. The absorptive power of a solution is indicated by that water-vapor pressure with which the solution will be in equilibrium. The moisture content of the leaving air can thus be controlled by adjusting the concentration of the solution.

The two solutions now used for absorption systems are calcium chloride and lithium chloride. The apparatus consists of a chamber through which the air flows and in which the solution is sprayed or flowed, over surfaces, so as to give a large contact area. The solution, of course, becomes dilute and must be reconcentrated. This is done by evaporating off the surplus water in a simple boiler that is part of the apparatus.

As the moisture is absorbed, its latent heat plus the "heat of mixing" is liberated and enters the air stream, also partly warming the solution. The latent heat is somewhat over 1000 B.t.u. per lb. of water absorbed; and the heat of mixing, at moderate concentrations, is 50 to 150 B.t.u. per lb.

Absorption systems are being developed and are available but are not as yet an important factor in air conditioning as measured by the number of units in use. They have the advantage of making use of heat in the process of reconcentration and therefore are of interest to the gas companies. They require, as does the adsorption method, a supply of water to cool the air after it passes through the absorber.

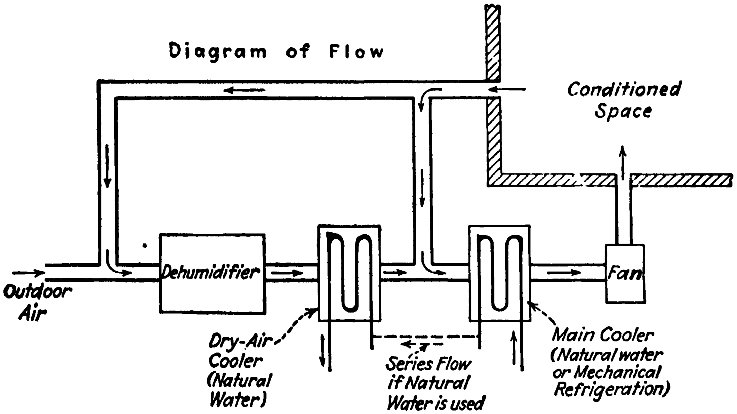
Diagram of Flow
Conditioned Space
Outdoor Air
Dehumidifier
Fan
Dry-Air Cooler (Natural Water)
Series Flow if Natural Water is used
Main Cooler (Natural water or Mechanical Refrigeration)

FIG. 284.—Diagrammatic arrangement of silica gel system.

341. The Silica Gel Method.—Adsorption is the phenomenon by which water is entrapped from the air by a porous substance. Adsorption is not fully understood but is a process of surface condensation.

In air conditioning, the adsorbent used is silica gel, a hard, glasslike substance that has the same chemical analysis as silica sand. When it has adsorbed its full capacity of water (though still dry to the touch), it can be reactivated by the application of heat. A silica gel system consists of a cylindrical basket of gel crushed to about the size of wheat and arranged so that air to be dehumidified and hot air for the reactivating process may be passed through the unit. The dehumidified air leaving the gel container is warm because of the latent heat of the water vapor that is released when the water is adsorbed and because of the heat of adsorption. Therefore, it is necessary to cool the air leaving the silica gel by means of water coils in order to reduce its temperature to a point suitable for introduction to the rooms.

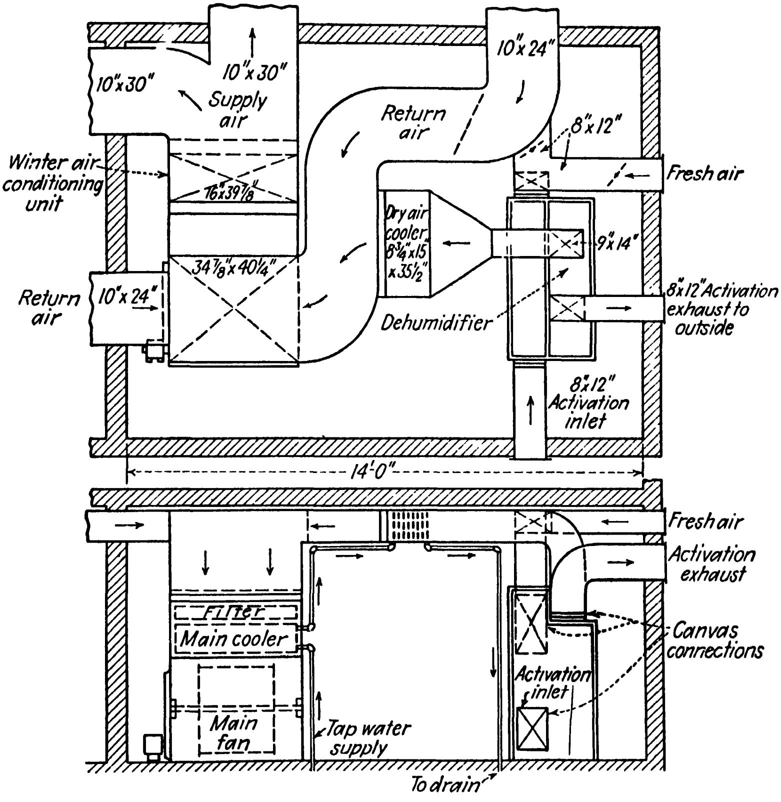

FIG. 285.—Installation of silica gel system. (*Bryant Heater Company.*)

Figure 284 shows one diagrammatic arrangement of a silica gel system, and Figs. 285 and 286 show the details of a self-contained rotary-type dehumidifier, as used in residences and other small installations. The unit illustrated in Fig. 286 is separated into two compartments with dividing seals located on the inside and outside of the drum. The drum rotates at the rate of approximately 1 revolution in 7 minutes. The portion of the drum in the right-hand compartment is drying air, while the section of the drum in the left-hand compartment is simultaneously being reconditioned.

As a particle of silica gel rotates from Z to X, it adsorbs water vapor until saturation is attained, and then, as it moves from X to Y, it is in contact with heated air from the gas burner. When the gel reaches Y, all of the water vapor has been evaporated into the activation air stream. It then enters segment $Y - Z$, where unheated air partially reduces its temperature preparatory to entering the adsorption compartment.

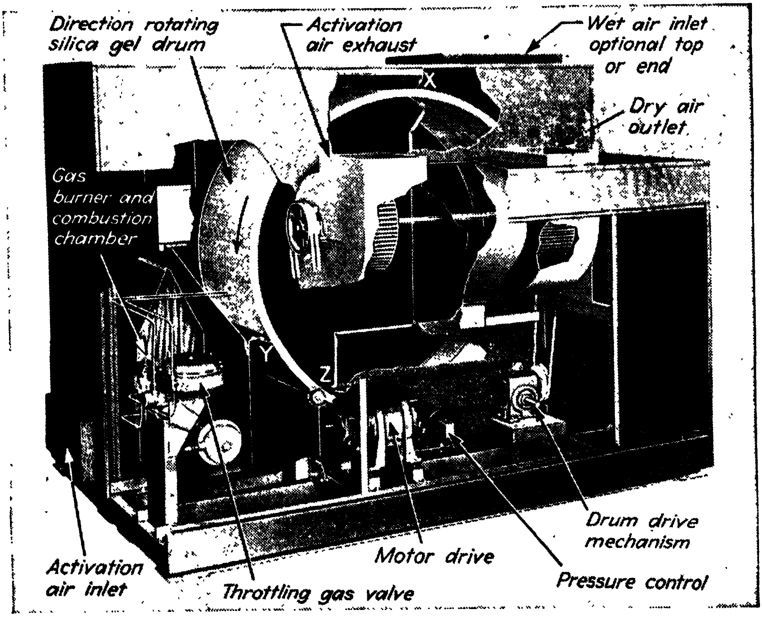

FIG. 286.—Cut-open view of silica gel unit. (*Bryant Heater Company.*)

Since the gel dehumidifies the air to a much lower moisture content than would be desirable in the rooms, only a portion of the return air from the rooms, mixed with some outdoor air if desired, is circulated through the gel bed, and this dry air after partial cooling in the aftercooler is mixed with the main stream of recirculating air. This mixture is then passed through the main cooler, where it is cooled to a few degrees above the incoming water supply before being delivered to the rooms.

To reactivate the gel, a gas or oil furnace is used, and the combustion gases are mixed with air to produce a resultant temperature of approximately 325 to 350° entering the gel bed. In some instances high-pressure steam is used to heat the reactivating air supply.

If cool water (70° or below) is not available, either mechanical refrigeration or evaporative cooling must be used to supplement the silica gel system. The mechanical refrigeration, since it is not used for dehumidifying, may operate at an evaporator temperature of 60 to 65°, and the power consumption is relatively small.

The use of evaporative cooling, following silica gel, is appropriate under certain conditions. Previous discussion has demonstrated how the dry-bulb temperature of the air can be lowered by the heat of evaporation being drawn from the sensible heat of the air. Although that method is not usually effective as the sole means of cooling, it is practicable when combined with a silica gel system because of the low relative humidity of the air supply that this system assures. When evaporative cooling is used with silica gel, a larger proportion of the air is dehumidified and is then partially rehumidified by passing it through a spray chamber in which the spray water is recirculated. The air can thus be cooled considerably below the temperature of the available supply water.

342. Field of the Silica Gel System.—It is evident that the silica gel system is essentially a dehumidifying rather than a cooling system. Its suitability for a particular application depends largely on whether the load is principally dehumidifying or principally cooling. If the former, this method, without employing evaporative cooling, is basically well suited. If, however, the requirement in any given case is principally the removal of sensible heat, the silica gel system is somewhat handicapped and must resort to evaporative cooling or be supplemented by a mechanical refrigeration system. The relation of the dehumidifying load and the cooling load will be more fully considered later.

Economically, the desirability of the silica gel system depends upon the cost of water and upon the cost of gas or oil if high-pressure steam is not available. The total water cost depends, in turn, upon the water temperature. All these factors must be evaluated and totaled for the case in question, including a consideration of the first cost of the equipment. The silica gel system is naturally favored by the gas industry because it can utilize gas.

In certain industrial applications where extremely low humidities are desired, silica gel is particularly suitable. To produce a very low humidity with mechanical refrigeration requires much reheating to bring the air back to a suitable dry-bulb temperature and is at best an expensive operation. Silica gel has been used to produce relative humidities as low as 0.5 per cent.

The future use of silica gel for comfort applications will be stimulated if it is generally agreed that for a given effective temperature a low humidity with a higher temperature is more desirable than a higher humidity and a lower temperature. At present there is little dependable knowledge concerning this point.

343. The Application of Refrigeration.—Methods used to remove heat from the air by refrigeration will now be considered.

The simplest method is to use a finned tube coil with direct expansion of the refrigerant into the coil. This method is employed in all small self-contained units and is used in central-fan systems of small size—up to about 50 tons capacity. The limit of size is a matter of legal restriction in many cities because of the possibility of leakage of the refrigerant into the air stream.

Where direct expansion is not feasible or not permissible, the refrigerating machine is used to chill water, and the water is circulated by a pump, either through a coil or through a spray chamber, which is simply an air washer equipped with an ample number of nozzles. The coil is preferred for simplicity of operation and the small space required, but the spray chamber provides greater accuracy of control. The reason for this is that the spray chamber (with a sufficient number of banks of nozzles) always delivers saturated air, and hence the dew-point method of control (see Art. 289, Chap. XVI), which is simple and accurate, is applicable. In industrial air conditioning, where exact control of humidity is usually necessary, the spray chamber is widely used. In comfort work the spray chamber has the advantage of permitting evaporative cooling to be used when conditions are favorable.

When a coil is used, the control is usually only a dry-bulb temperature control, and the dew point of the air will vary through an appreciable range. In comfort cooling this is quite permissible.

344. Cooling Coils.—Cooling coils are usually constructed of copper alloys and have fins, similar to those used for heating, as illustrated in Fig. 245, Chap. XVI. In cooling work the tubes are always horizontal and the fins vertical, so as to permit the condensation to drain readily from the coil surface. The arrangement of coils with respect to air flow and flow of cooling fluid is subject to many variations.

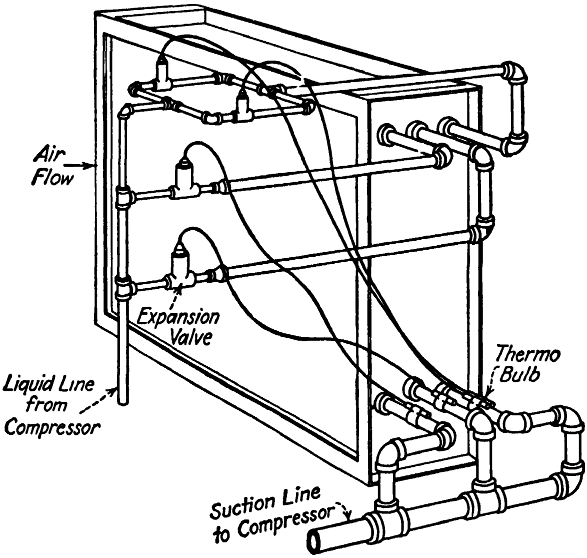

FIG. 287.—Cooling coil arranged for direct expansion.

When the coils are located in the air stream to be cooled, and the refrigerant is expanded within the coil, direct expansion takes place. One form of direct-expansion coil arrangement is shown in Fig. 287. Liquid refrigerant is admitted to the cooling coil through expansion valves, which are controlled by means of a thermostatic bulb clamped to the suction line as it leaves the coil. When the refrigerant enters the region of low pressure prevailing within the coil, enough liquid flashes into vapor to cool the remaining liquid to the prevailing saturation temperature. The liquid boils when the temperature of the fluid (air or water) surrounding the coils is above the refrigerant boiling point, and it is thus converted to a vapor by absorbing its latent heat of vaporization from the fluid. The vapor is then returned through the suction line to the compressor to repeat the cycle.

Sometimes two expansion valves in parallel are required to obtain the necessary capacity. To permit the use of fewer and larger expansion valves, various forms of distributing headers have been devised to supply every tube in each circuit of a coil with an even and equal amount of refrigerant.

Parallel flow of refrigerant with respect to air flow is generally adopted, as the entire coil surface temperature is then fairly uniform. Sometimes the refrigerant circuit is brought back through a single row of tubing on the entering air side to take advantage of warmest air to superheat the refrigerant vapor. Coil performance and selection are discussed in Chap. XIX.

Cooling coils for use with chilled water are constructed differently. To reduce the amount of water circulated and the amount of coil surface, is is desirable to have a counterflow arrangement. This means that the water and air generally flow in opposite directions, so that the incoming cold water enters the face of the coil from which the air is leaving, and the outgoing water leaves the coil from the face at which the air is entering. The coils are in series with respect to water flow. The physical construction of the cooling units for these two purposes is therefore somewhat different.

Cooling coils fabricated of bare pipe may be submerged in open tanks for cooling water or brine, or they may be arranged in a closed system with shell-and-tube construction. In the latter arrangement the refrigerant is generally in the tubes and water or brine are in the shell, with thermostatic expansion valve control.

When comparing direct-expansion and water-cooled systems it must be realized that the former will operate at a higher suction pressure because one less heat transfer is required. Therefore direct-expansion systems result in lower first cost and operating expense. The refrigerant used must be nontoxic.

345. Expansion Valves.—A typical expansion valve as used on cooling coils is shown in Fig. 288. The thermostatic bulb is attached to the suction line from the coil as shown in Fig. 287. An increase of coil temperature causes the valve to open wider, admitting more refrigerant to the coil. Opposing this force is the pressure of the refrigerant on the low-pressure side of the valve, which is exerted against the lower bellows and tends to close the valve.

346. Condensers.—The heat released from a compression-type refrigerating system will be about 15,000 B.t.u. per hr. per ton, of which 12,000 B.t.u. per hr. represents the heat absorbed in the cooling coil and 3,000 B.t.u. per hr. is the heat equivalent of electric input to the compressor motor. Condensing-water quantities vary from 2 gal. per min. per ton of refrigeration for compression systems to as high as 5 gal. per min. for absorption systems, depending on the unit and operating conditions.

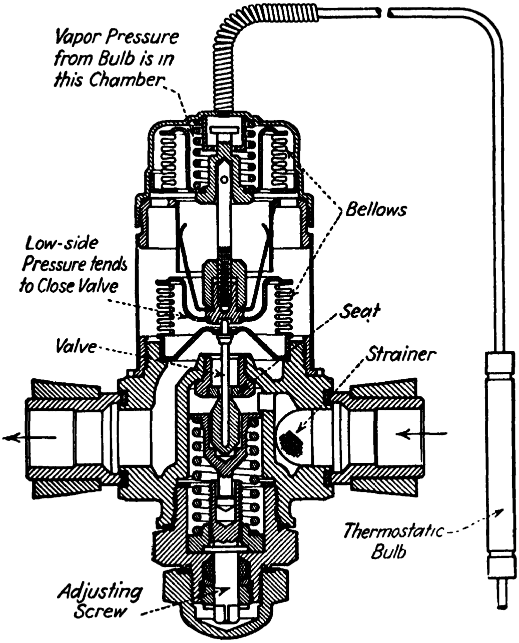
Vapor Pressure from Bulb is in this Chamber
Bellows
Low-side Pressure tends to Close Valve
Seat
Valve
Strainer
Thermostatic Bulb
Adjusting Screw

FIG. 288.—Expansion valve. (*Detroit Lubricator Company.*)

To dissipate the required heat release three general types of condensers are used, as follows:

Air-cooled condensers.
Water-cooled condensers.
  a. With water going to waste.
  b. With cooling tower.
Evaporative condenser.

The **air-cooled condenser** is simply a finned tube coil over which air is blown by a fan. The temperature of the cooling air is naturally higher in nearly any case than that of water which might be used, and the power consumption of the refrigerating machine is therefore relatively high. Air condensers are used in self-contained room units; in small machines, up to 3 hp.,

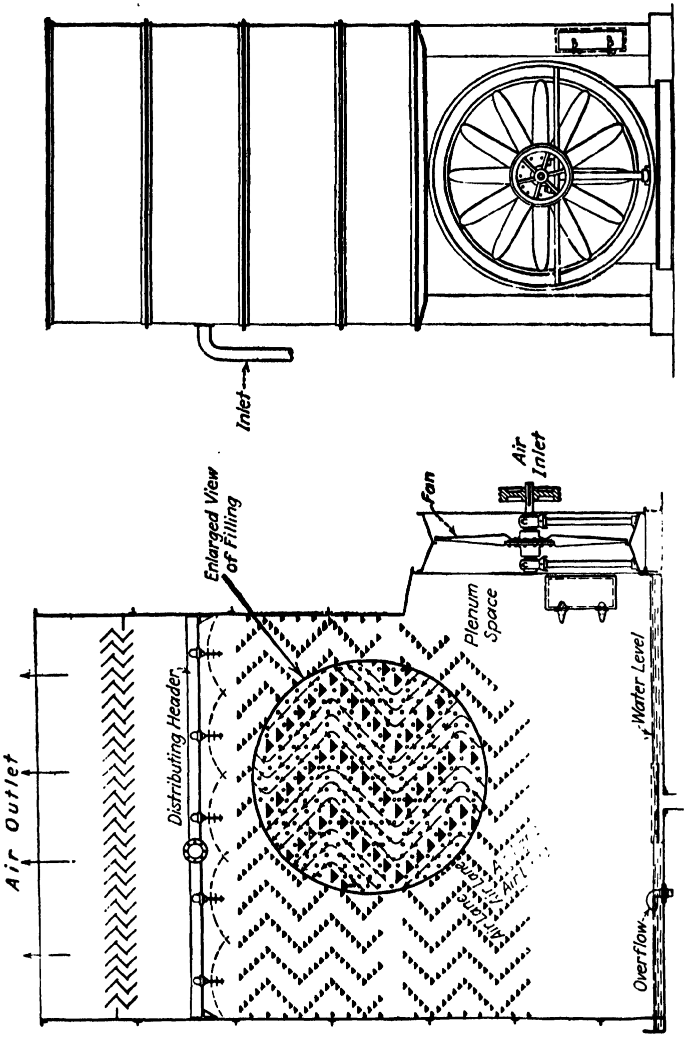

FIG. 289.—Cooling tower. (*The Cooling Tower Company, Inc.*)

where simplicity and low first cost are desired; and in railway air-conditioning systems.

Where city water is available at a reasonable cost and at a temperature not exceeding about 75°, the **water-cooled condenser,** with the water going to waste, is usually the most practicable method. However, the cost or high temperature of city water, the prohibition of its use for such purposes in large cities, or the inadequacy of the sewers has led to the use of the cooling tower and the evaporative condenser. In general, they are justified from an economic standpoint only when the cost of water is about $1 or more per thousand cubic feet.

Generally this type of condenser is of the shell-and-tube type, with water circulated through the tubes and the condensing refrigerant vapor occupying the shell. As condenser temperatures are high, scaling and mineral deposit[1] on the water side of tube surfaces is likely to be a source of trouble. For this reason, shell-and-tube condensers with straight tubes and removable heads are preferred, as they permit easy cleaning of the tubes.

In the **cooling tower** the water is sprayed or sprinkled into a tower filled with latticework, usually of wood, and drips down through the upflowing stream of air. The so-called "atmospheric tower" relies on the wind, plus a certain amount of stack effect, to circulate the air. The fan tower, which is the type principally used in air conditioning, consists of a metal boxlike structure with a propeller fan located at either the top or the bottom, forcing the air upward. One design is shown in Fig. 289.

The maximum cooling effect that could theoretically be obtained is limited by the wet-bulb temperature of the air. The efficiency of cooling is

$$E = \frac{t_1 - t_2}{t_1 - t} \qquad (2)$$

where t = wet-bulb temperature of the air.

t_1 = temperature of the water entering tower.

t_2 = temperature of the water leaving tower.

The efficiency usually ranges between 70 and 75 per cent. The water is usually cooled through a range of 10 to 15°, and its leaving temperature is obviously a function of the outside wet-bulb temperature.

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[1] "Survey of Water Problems Encountered with Summer Air Conditioning Equipment," *Am. Gas Assoc. Research Bull.* No. 31, 1944.

When space is limited, a spray chamber, designed like a conventional air washer, is sometimes used for condensing-water cooling.

Some water is lost from a cooling tower as the result of evaporation and of drift. The total, for a system using a refrigerant such as F-12, is about 0.06 gal. per min. per ton of refrigeration. With no cooling tower, the amount of city water used and wasted is from 1½ to 2 gal. per min. per ton, depending upon the water temperature. The cooling tower is generally assumed to give a 90 to 95 per cent saving of water.

347. The Evaporative Condenser.—Another method of conserving water is to use an evaporative condenser. This consists of a finned-tube condenser enclosed in a housing, a set of water sprays supplied by a pump, and a fan to circulate air through the housing. Figure 290 shows a typical design.

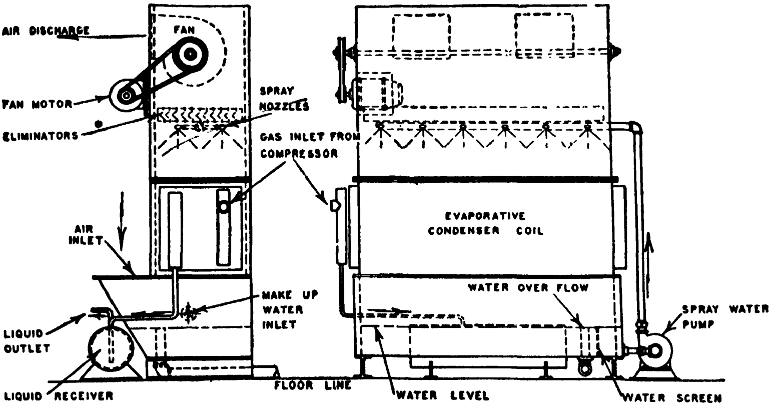

FIG. 290.—Evaporative condenser.

The evaporative condenser is virtually a combination of a cooling tower with a water-cooled condenser. It has a high rate of heat transfer, is often cheaper to install, and has the advantages of the cooling tower in saving water, although the water consumption due to evaporation is somewhat higher. The chief disadvantage is the difficulty of finding a suitable location in the building where the air-supply connections and the refrigerant connections are reasonably short.

Ratings for evaporative condensers are customarily based on net useful refrigerating effect, instead of total heat dissipation accomplished, because it is necessary to correlate condenser

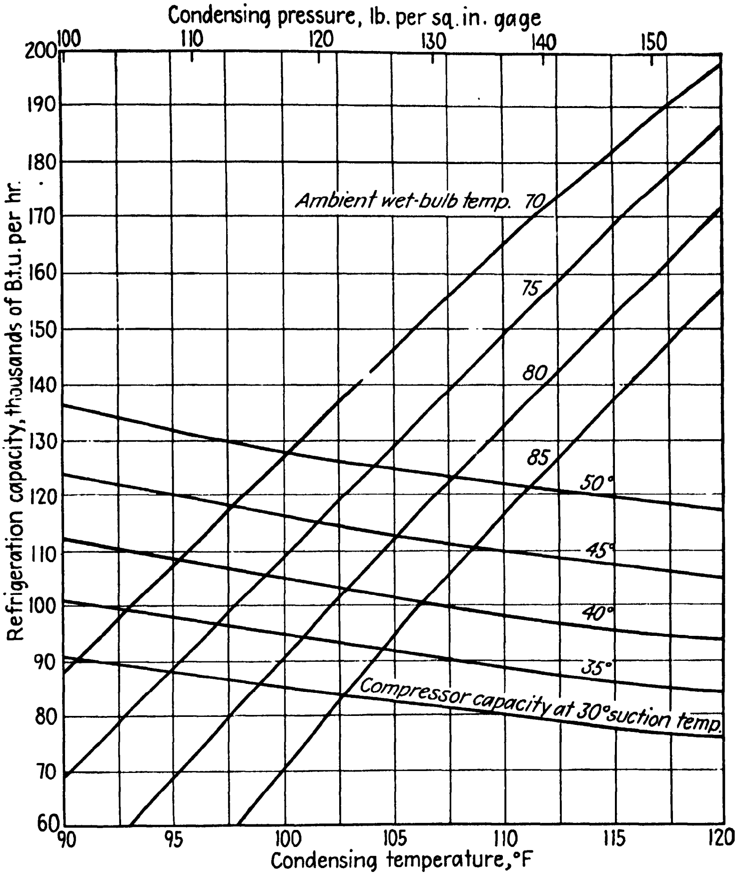

Fig. 291.—Typical evaporative condenser-compressor rating curves.

capacity with that of a matched compresser. Motor loads for various capacity ratings are given in Table 89.

Table 89.—Motor Loads for Various Evaporative-condenser Capacities

Capacity		Water pump	Fan hp.	
B.t.u. per hr.	Tons		Without ducts	With ducts
60,000	5	⅙-¼	½-1	1-1½
240,000	20	⅙-½	1½-3	3-5
600,000	50	¾-1	10-15	10-15

Choice of maximum design wet-bulb temperature is more critical for sizing an evaporative condenser than for establishing the capacity of an air-conditioning system. With the latter, the only effect is to raise the indoor temperature or humidity. In the case of evaporative condensers, a rise in wet-bulb temperature results in an immediate reduction of condenser capacity and consequently an increase of condensing pressure and temperature.

Typical evaporative condenser-compressor capacity curves are illustrated in Fig. 291. It is apparent that a given change in wet-bulb temperature affects capacity to about the same extent as an equal change in condensing temperature. In general, compressor capacity increases from 0.6 to 0.8 per cent for each 1° reduction in condensing temperature. Low condensing temperatures result in economical operation, because the power input to the compressor is reduced, but at the same time it should be recognized that a large-size evaporative condenser will be required.

The evaporative condenser is used for moderate-size installations (25 to 100 tons) and has a definite field. The trend is toward the restriction of the use of city water in large quantities, and both the evaporative condenser and the cooling tower are destined to be important.

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Problems

1. It is desired to use ice for cooling a building that has a total calculated cooling and dehumidifying load of 50 tons of refrigeration. During the most severe day the load on the system is expected to be the equivalent of 5 hr. of full-load operation. The temperature of discharge of the ice water to the sewer is 50°.

(*a*) How many cubic feet of ice must be provided for in the ice tank for one day's use? (*b*) How much will the ice cost for 20 days of full-load operation at $5 per ton? (c) If the temperature rise in the cooling coil is 8°, how much water must be circulated in gallons per minute?

2. The over-all coefficient of performance of a refrigerating plant, reckoned from the electrical input to the motor, is 3.2. The total refrigerating load is 17 tons. What will be the operating cost per hour at full load if electricity costs 2½ cents per kilowatt-hour?

3. In a steam-jet system what volume of vapor must be handled per minute by the jet compressor if the total load is 12 tons and the water leaves the cooling coil of the air-conditioning system at 55°?

4. What will be the seasonal operating cost for power plus water for a 50-ton refrigeration system which operates under the following conditions?

220 hr. at 100 per cent load 1.2 hp. per ton.
360 hr. at 75 per cent load 1.25 hp. per ton.
410 hr. at 50 per cent load 1.3 hp. per ton.
180 hr. at 25 per cent load 1.35 hp. per ton.

The average water requirements are 1.7 g.p.m. per ton. Water costs $0.60 per thousand cubic feet, and electricity costs $0.02 per kilowatt-hour. Motor efficiency 90 per cent at full load, 87 per cent at three-quarters load, 84 per cent at half load, and 80 per cent at one-quarter load.

5. A cooling system using water in the cooling coils requires 85 gal. per min. The temperature rise of the water is 6°. Make the necessary assumptions as to the performance of the refrigerating machine, and calculate the kilowatt-hours required for an 8-hr. period of operation.

CHAPTER XIX

THE COOLING AND DEHUMIDIFYING LOAD

348. General Considerations.—This chapter deals with the important problems of calculating the cooling and dehumidifying load, determining the air quantities to be circulated, and selecting the coils or spray chamber.

The calculating of the cooling and dehumidifying load is important because the capacity of the cooling system as a whole and the amount of cooling provided for each room are based on that calculation. In general the cooling-load figures have the same significance in cooling-system design as do the heat-loss figures in heating-system design. The methods of calculation, however, differ in many respects. There is also the dehumidifying load to be considered in the cooling-system design.

In most parts of the country the outdoor temperatures at night fall to the range of comfortable conditions, so that the heat flow into a building structure is decidedly intermittent and often is alternating in character. Heat may flow out of the structure at night and into the structure in the daytime. The *thermal capacity* of the outer walls and roof also has a marked effect by causing a time lag in the penetration of heat to the room.

Another important item in the cooling load is the effect of radiant heat from the sun. It causes an increased heat flow through walls and roofs by their absorption of radiant heat; and it penetrates through windows on the sunny sides of the building unless they are externally shaded.

The shifting of the sun effect during the course of the day has a marked influence on the cooling load. In many buildings the total cooling load for which a central system must be designed is considerably less than the sum of the maximum cooling loads of the individual rooms.

The effect of artificial lighting is often important and must be taken into account, though not necessarily to the full equivalent of the electricity consumed.

The dehumidifying load is caused by the infiltration of moist air from out-of-doors, by the moisture given off by human beings, and by the moisture arising from cooking and other processes.

The calculation of the cooling load is in many respects more empirical and less accurate, in the present state of the art, than is the calculation of the winter heating load, and the nature of the problem makes it less amenable to exact mathematical treatment.

In further comparison of the cooling load with the winter heating load, it may be pointed out that the effects of the sun, lights, people, and heat-emitting processes tend to *reduce* the amount of heat required in winter, but they cannot be taken advantage of because they are not assuredly present at all times; whereas these same factors in summer tend to *increase* the cooling load and must be assumed to be effective at the time of the maximum cooling load.

The various items of the cooling and dehumidifying load may be listed as follows:

Cooling Load:

1. Transmission through building walls, roof, and windows.
2. Sun radiation through windows.
3. Heat in outside air entering by infiltration or ventilation.
4. Heat from occupants.
5. Heat from artificial lighting.
6. Heat from cooking and other processes.

Dehumidifying Load:

1. Moisture in outside air entering by infiltration or ventilation.
2. Moisture from occupants.
3. Moisture from cooking and other processes.

349. Design Temperatures.—The outdoor conditions to be assumed in the calculation of the cooling and dehumidifying load should not be the highest recorded temperature, which usually exists for only a few hours during the year. They should be chosen as conditions which may be exceeded for a few hours during the summer. In every large city a certain design dry-bulb and wet-bulb temperature, which are readily ascertained, have become established and generally adopted. Table 90 gives these temperatures for a few cities.

TABLE 90.—DESIGN DRY-BULB AND WET-BULB TEMPERATURES

Location	Design temperatures, °F.	
	Dry-bulb	Wet-bulb
New York City	95	75
Chicago	95	75
Philadelphia	95	78
Detroit	95	75
Cleveland	95	75
Boston	92	75
Pittsburgh	95	75
St. Louis	95	78
Washington	95	78
New Orleans	95	79
San Francisco	90	65
Dallas	100	78
Jacksonville	95	78

350. Transmission through Walls.—The transmission of heat into a building by conduction through the walls, roof, and windows is not simply the reverse of the process of heat loss in winter. It is complicated by two important factors.

The first of these is evident from a study of Fig. 292, which shows the relation between the indoor and outdoor temperature in Detroit for three winter days and for three summer days. In winter the outdoor temperature is continually far below the indoor temperature, and there is always a flow of heat outward. The radiant heat from the sun striking the walls and roof raises the outer wall-surface temperature and decreases the heat flow, but this cannot be taken into account in the calculation of the maximum heating load, because the sun cannot be assumed to be shining in the coldest days. Thus in winter, although there is some fluctuation of heat flow and some effect of the thermal capacity of the structure, it is quite permissible to treat the problem as one of steady heat flow.

In summer, as Fig. 292 shows, the outdoor temperature in the latitude of Detroit is well above the indoor temperature during the day but at night drops down to or below the indoor temperature. This causes a reversal of heat flow, and there is a definite cyclic effect due to changes in the outdoor air temperature, which is further emphasized by the effect of the sun.

The sun effect must be taken into account in cooling-load calculations because the sun will be shining during the days of maximum cooling load. The sun heat on the roof and on walls exposed to the sun raises the outside surface temperature above the outdoor air temperature, to an amount depending largely upon the color and character of the surface. Thus because of these two factors—the daily fluctuation of outdoor temperature and the sun effect—the phenomenon of heat flow through walls and roofs in summer is much more complex than in winter, and the cooling-load calculation must take this into account.

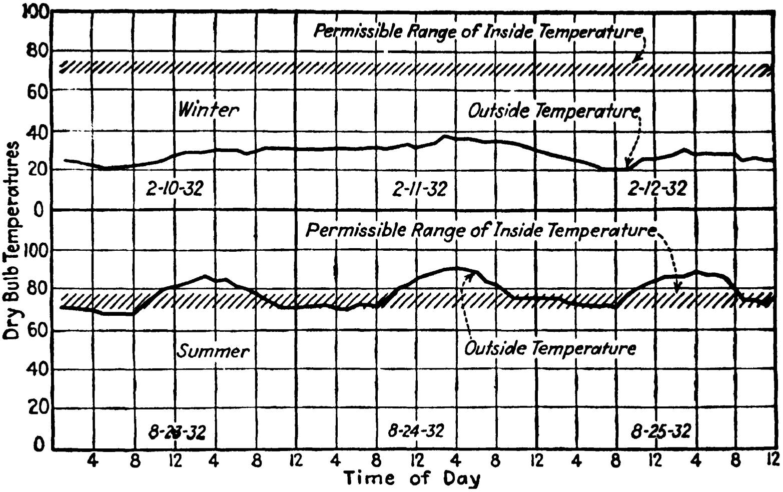

FIG. 292.—Relation of inside and outside temperatures in heating and in cooling buildings (Detroit).

As an extreme example of the effect of the thermal capacity of a wall, note Fig. 293, which shows the actual heat flow at the outer and inner surfaces of a heavy masonry wall facing nearly south and exposed to the sun. The heat flow was measured by Nicholls meters (Art. 33, Chap. II) placed upon the inner and outer wall surfaces. The heat flow into the wall at the outside surface reached a high peak in the early afternoon. At night the direction of heat flow from the outer surface reversed, and outward heat flow continued until the wall was again warmed by the sun's rays in the morning. The heat flow into every room from the inner surface of the wall was irregular and very small in amount. It showed no decided peak but was highest during the early part of the cooling period, when the reduction of room temperature naturally increased the heat flow from the wall surface to the room air. This wall was of unusually heavy construction[1] and the case is rather extreme, but it strikingly illustrates the point.

351. Solar Radiation.—Before proceeding with the calculation of wall transmission it is desirable to understand how the actual radiant heat from the sun varies throughout the day. There are several modifying factors such as the latitude, the season of the year (both of which govern the solar elevation), the orientation and angle of the intercepting surface, and the condition of the atmosphere.

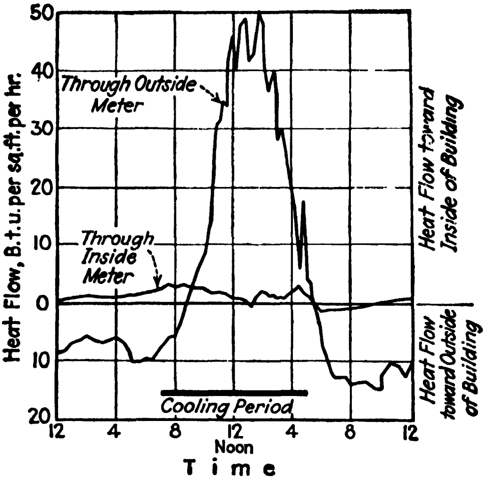

FIG. 293.—Heat-flow conditions at inner and outer surfaces of a wall of a cooled building.[2]

The intensity of solar radiation is easily measured with the pyrheliometer, which gives the amount of heat that strikes a plane located at an angle normal to the sun's rays. The actual heat per square foot of window area at any instant is this figure for a normal plane, multiplied by the sine of the angle between the sun's rays and the surface. This angle varies with the time of day, season, latitude, and the orientation of the window.

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[1] Plaster, 1¼ in.; hollow tile, 3¾ in.; air space, 5¼ in.; hollow tile, 3¾ in.; brick, 9 in.

[2] From WALKER, SANFORD, and WELLS, "Field Studies of Office Building Cooling," *Trans. A.S.H. & V E.*, 1932.

In addition to the actual radiant heat from the sun, a certain amount of diffuse radiation, from reflection and refraction, is received from all parts of the sky. This scattered or sky radiation was found to be up to 20 per cent of the total radiation measured in some localities.

Figure 294 shows the intensity of solar radiation (direct plus sky) for August 1 in the latitude of Pittsburgh for surfaces oriented in various directions. Note that for a surface normal to the sun's rays the maximum intensity is approximately 310 B.t.u. per sq. ft. per hr. The intensity is considerably decreased by a hazy or smoky atmosphere. The effect of these factors has been studied and the results have been compiled in complete sets of tables,[2] but for ordinary purposes, as will be shown later, it is permissible to use approximate figures.

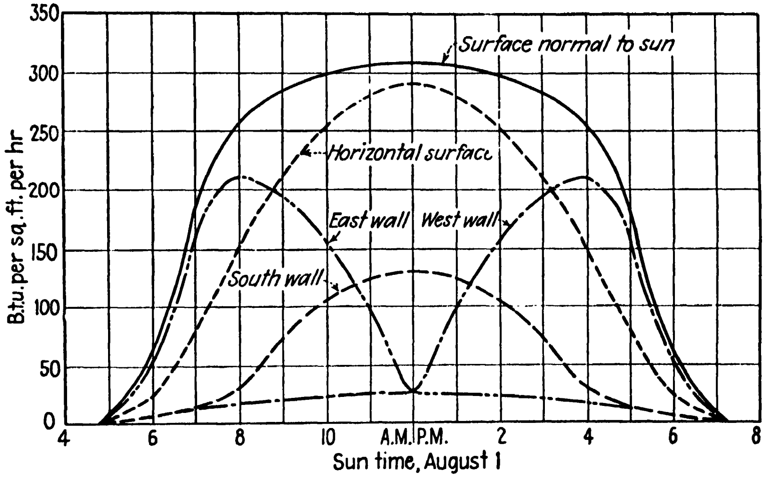

FIG. 294.—Solar intensity normal to sun on horizontal surface and on walls for August 1 at 40 deg. N. Lat.[1]

352. Intermittent Heat Flow through a Wall.—The process of heat flow through a wall or roof exposed to the sun is quite complex. The variation of the outdoor air temperature alone causes heat to flow into the outer surface of the wall during the day and

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[1] HOUGHTEN, GUTBERLET, and BLACKSHAW, "Studies of Solar Radiation through Bare and Shaded Windows," *Trans. A.S.H. & V.E.*, 1934.

[2] HENDRICKSON and WALKER, "Summer Cooling for Comfort as Affected by Solar Radiation," *Heating and Ventilating*, November, 1932; "The Determination of Sun Effect on Summer Cooling Loads," *Idem*, June, 1933.

(in most latitudes) outward at night. A wave of heat, so to speak, of mild intensity is initiated and progresses through the wall. Superimposed on this is a much more intense wave of shorter duration caused by the radiant heat from the sun. Not all the heat striking the outer wall surface goes inward, however. One component is reradiated to surrounding surfaces, and another portion is carried off by convection currents of the outdoor air, which is cooler than the wall surface. But a wave passes through the wall, having a magnitude and speed governed by the conductance of the wall material and by its thermal capacity.

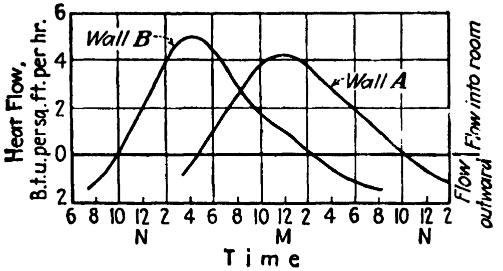

FIG. 295.—Heat flow into a building from inner side of two walls.

Wall *A*:

13-in. brick, 1-in. plaster.

$U = 0.34$. Thermal capacity per deg. change in temperature = 25.3 B.t.u. per sq. ft.

Wall *B*:

Shingles, sheathing, 2- by 4-in. studs, 1-in. blanket insulation, lath, and plaster.

$U = 0.12$. Thermal capacity per deg. change in temperature = 2.8 B.t.u. per sq. ft.

This lag may be as much as 3 hr. for a 6-in. concrete wall and 10 hr. for a 22-in. brick and tile wall. Of two walls having the same over-all *conductance* but different thermal *capacities*, the one having the greater capacity will usually throw much less actual load upon the cooling system.[1]

Thus from the different walls of a building there will be components of the cooling load differing in magnitude and in the time at which they are at a maximum—that is, when the crest of the heat wave reaches the inner wall surface.

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[1] See HOUGHTEN and others, "Heat Transmission as Influenced by Heat Capacity and Solar Radiation," *Trans. A.S.H. & V.E.*, 1932.

Figure 295, which shows the heat flow through two actual walls facing north, illustrates the foregoing. Wall A has a higher heat transmission and a much higher thermal capacity than wall B. The amount of heat passing into the room from wall A is less, at its maximum, than that from wall B, and its peak does not occur until midnight, whereas the peak for wall B occurs at 4 P.M. If the cooling period is from, say, 8 A.M. to 5 P.M., the heat gain through wall A would not enter into the actual cooling load except for such portions as enter the room the following morning.

Although several complex mathematical solutions[1] have been evolved for calculating the heat gain through walls, at the present time there is no generally accepted practical method of predicting this component of the cooling load. An approximate method[2] for estimating the heat flow through walls utilizes the term **sol-air temperature.** This is an equivalent outdoor-air temperature which, in contact with a shaded surface, would give the same rate of heat transfer and the same temperature distribution through that material as exists with the actual outdoor-air temperature and incident solar radiation.

$$t_e = t_o + \frac{aI}{h} \tag{1}$$

where t_e = sol-air temperature, deg. F.
t_o = outdoor-air temperature, deg. F.
a = absorptivity of the outside surface for solar radiation.
I = intensity of incident solar radiation, B.t.u. per hr. per sq. ft.
h = surface coefficient of heat transfer between outdoor air and surface of material, B.t.u. per sq. ft. per deg. F.

Example.—Assume the outdoor-air temperature in the shade to be 95° and the intensity of solar radiation incident upon a concrete wall to be 100 B.t.u. per hr. per sq. ft. For a surface absorptivity of 0.7 and an outside film coefficient of 5.0, which corresponds to 10 m.p.h. wind velocity (from Fig. 8, Chap. II), the sol-air temperature is

$$t_e = 95 + \frac{0.7 \times 100}{5} = 109^\circ$$

In other words, air at a dry-bulb temperature of 109° in contact with a shaded wall is equivalent to the combined effects of actual air temperature and solar radiation.

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[1] ALFORD, RYAN, and URBAN, "Effect of Heat Storage and Variation in Outdoor Temperature and Solar Intensity on Heat transfer through Walls," *Trans. A.S.H. & V.E.*, 1939.

[2] MACKEY and WRIGHT, "Periodic Heat Flow—Homogeneous Walls or Roofs," *Trans. A.S.H. & V.E.*, 1944.

Some further checking and experimental work is required before the concept of sol-air temperature can be generally accepted, but it does seem to offer a promising approach to solving one of the complex factors involved in calculating the variable heat flow through walls.

Until a sound method based on additional laboratory or field data can be developed, it is recommended that the following empirical approach be utilized. This method[1] may not be in close agreement with actual field conditions, but is used as an interim procedure pending the obtaining of actual data on various walls.

The heat gain through a wall may be expressed by the equation

$$H = H_t + H_R \qquad (2)$$

$$H_t = AU(t_o - t) \qquad (3)$$

where H_t = heat gain due to air temperature difference.
A = area of wall, square feet.
U = coefficient of heat transmission, B.t.u. per sq. ft. per hr. per degree difference in temperature between room air and outdoor air.
t_o = outdoor air temperature, deg. F.
t = room air temperature, deg. F.

and

$$H_R = AFaI \qquad (4)$$

where H_R = heat gain due to sun radiation.
A = area of wall or roof, sq. ft.
F = portion, expressed as a decimal, of the absorbed solar radiation that is transmitted to the inside.
a = portion, expressed as a decimal, of the impinging solar radiation that is absorbed by the wall surface.
I = actual intensity of the solar radiation striking the surface, B.t.u. per hr. per sq. ft.

The values of F are related to the transmission coefficient of the wall, and the relationship[2] is approximately

$$F = 0.23U \qquad (5)$$

---

[1] This method is essentially that proposed by Faust, Levine, and Urban in "A Rational Heat Gain Method for the Determination of Air Conditioning Cooling Loads," *Trans. A.S.H. & V.E.*, 1935.

[2] Originally shown as a curve in the paper in Ref. 1 above.

and Eq. (4) becomes

$$H_R = 0.23AUaI \tag{6}$$

The values of a, the absorption coefficient, are given in Table 93. Those of I vary with the latitude and season. For a latitude of 40 deg.[1] on August 1, the values are given in Table 92.

Equation (6) obviously does not take into account the time lag in the passage of the heat wave through the wall; it gives only the magnitude of the heat flow to the inner wall for given values of I. The time lag must be estimated from a small amount of data that have been obtained on certain materials as shown in Table 94. It is necessary to select from Table 94 a value for a wall construction somewhat similar to the wall in question.

By the use of Eqs. (2), (3), and (6) it is thus possible to construct an approximate time curve of the heat passing into the room. For walls not exposed to the sun, H_R is zero, and the heat flow is represented by H_t.

Example.—Given a south wall, 4 in. brick, 8 in. hollow tile, ½ in. plaster on 1 in. rigid insulation; $U = 0.14$. Latitude 40 deg., date August 1. Outside temperature 95°. Inside temperature 78°. Outside wall surface medium dark. Calculate heat gain per square foot from noon to 10 P.M.

TABLE 91.—CALCULATION OF HEAT GAIN FOR EXAMPLE

Actual sun time	I from Table 92	H_R	H_t	H	Time heat enters room
6:00 A.M.	4.5	0.10	2.4	2.50	12:00 noon
7:00	11	0.25	2.4	2.65	1:00 P.M.
8:00	29	0.65	2.4	3.05	2:00
9:00	74	1.66	2.4	4.06	3:00
10:00	103	2.32	2.4	4.72	4:00
11:00	124	2.79	2.4	5.19	5:00
12:00 M	128	2.88	2.4	5.28	6:00
1:00 P.M.	124	2.79	2.4	5.19	7:00
2:00	103	2.32	2.4	4.72	8:00
3:00	74	1.66	2.4	4.06	9:00
4:00	29	0.65	2.4	3.05	10:00

$$H_t = 0.14(95 - 78) = 2.4 \text{ B.t.u. per sq. ft. per hr.}$$
$$H_R = 0.23 \times 0.14 \times 0.7 \times I.$$
$$H_R = 0.0225I.$$

---

[1] This latitude passes near Philadelphia, Pittsburgh, Columbus, Indianapolis, and Springfield, Ill. For other latitudes see A.S.H. & V.E. "Guide," 1945.

From Table 92 take values of I, and construct Table 91.

Assume the time lag to be 6 hr. This is done by inspection from Table 94. The wall in question has a thermal capacity and a conductivity somewhat like the 4-in. iron and cork wall in the table.

In the foregoing example the outside temperature is assumed to be constant throughout the period for which the calculation is made. If a variation in outdoor temperature is assumed, the variation in heat flow will be greater, but the difference will not be important in most cases.

TABLE 92.—SOLAR RADIATION (DIRECT PLUS SKY) IMPINGING AGAINST WALLS HAVING SEVERAL ORIENTATIONS AND A HORIZONTAL SURFACE[1]
(For 40 deg. north latitude on August 1)

Sun time	Intensity of solar radiation, B.t.u. per sq. ft. per hr.							
	North-east	East	South-east	South	South-west	West	North-west	Horizontal surface
4:50 A.M.	0	0	0	0	0	0	0	0
5:00	5	6	4	2.5	2.5	2.5	2.5	5
6:00	49	56	32	4.5	4.5	4.5	4.5	20
7:00	123	162	109	11	11	11	11	85
8:00	137	211	166	29	17	17	17	160
9:00	102	195	181	74	21	21	21	212
10:00	54	152	171	103	23.5	23.5	23.5	244
11:00	28	94	144	124	41	25.5	25.5	281
12:00 M	26	26	98	128	98	26	26	290
1:00 P.M.	25.5	25.5	41	124	144	94	28	281
2:00	23.5	23.5	23.5	103	171	152	54	244
3:00	21	21	21	74	181	195	102	212
4:00	17	17	17	29	166	211	137	160
5:00	11	11	11	11	109	162	123	85
6:00	4.5	4.5	4.5	4.5	32	56	49	20
7:00	2.5	2.5	2.5	2.5	4	6	5	5
7:10	0	0	0	0	0	0	0	0

The heat gain through walls is usually a minor part of the total cooling load, and the foregoing method therefore does not usually result in serious over-all error. In an important installation where the wall or roof heat gain is a major item, a more thorough study should be made. In any case the student should keep in touch with current research on the subject, which will eventually lead to a more proved method.

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[1] From Chap. 7, A.S.H. & V.E. "Heating, Ventilating, Air Conditioning Guide," 1945.

The proper method for determining the total cooling load for individual rooms and for the building as a whole, in the case of a large or important installation, is to plot or tabulate the heat gain for the hours of the cooling period, superimposing the different components to determine the maximum. There is a diversity in the times of the peaks of the various components that make up the load in each room, and there is a still more important diversity in the times of the peaks for the various rooms or exposures. Because of this diversity, the peak load for the entire building, upon which the size of the apparatus is

TABLE 93.—SOLAR ABSORPTION COEFFICIENTS FOR DIFFERENT BUILDING MATERIALS[1]

Surface Material	Absorption Coefficient (*a*)
Very light-colored surfaces:	
White stone	0.4
Very light colored cement	
White or light cream-colored paint	
Medium-dark surfaces:	
Asbestos shingles	0.7
Unpainted wood	
Brown stone	
Brick and red tile	
Dark-colored cement	
Stucco	
Red, green, or gray paint	
Very dark-colored surfaces:	
Slate roofing	0.9
Tar roofing materials	
Very dark paints	

[1] From *Trans. A.S.H. & V.E.*, vol. 41, p. 341, 1935.

TABLE 94.—TIME LAG IN TRANSMISSION OF SOLAR RADIATION THROUGH WALLS AND ROOFS[1]

Type and Thickness of Wall or Roof	Time Lag, Hr.
2-in. pine	1½
6-in. concrete	3
4-in. gypsum	2½
3-in. concrete and 1-in. cork	2
2-in. iron and cork (equivalent to ¾-in. concrete and 2.15-in. cork)	2½
4-in. iron and cork (equivalent to 5½-in. concrete and 1.94-in. cork)	7¼
8-in. iron and cork (equivalent to 16-in. concrete and 1.53-in. cork)	19
22-in. brick and tile wall	10

[1] From Chap. 8, A.S.H. & V.E. "Heating, Ventilating, Air Conditioning Guide," 1940.

based, is far less than the sum of the individual peaks of the various rooms.

353. Sun Radiation through Windows.—The effect of the sun through unshaded windows requires special consideration. The process is peculiar in many respects, and the effect on the cooling load in some buildings is important.

Ordinary window glass is rather transparent to sun radiation, and the entering heat, if it strikes furniture or other objects in the room, enters directly into the cooling load. It cannot be reradiated outward through the glass even by a shade placed close to the window, because the *low-temperature* radiation (that is, the longer wave lengths) has been found not to pass through glass to any appreciable extent. Therefore, the only effective way of shutting out solar radiation is to use *outside* shades or awnings.

Tests have shown that ordinary window glass absorbs as much as 16.7 per cent of the heat striking it perpendicularly and that two ¼-in. panes absorb 37.5 per cent. Special glasses are available that absorb larger amounts without proportionately decreasing the transmission of light.

The heat intercepted by the glass raises the glass temperature, and heat passes from the glass to the room air by convection. If the outdoor air is at a higher temperature than the glass, heat will flow into the glass by convection and thence to the room; if the outside air is cooler than the glass, part of the radiation intercepted by the glass will pass to the outside air. It would be impracticable to take all these factors into account, and it has been found that on the average the heat entering through a bare window from all sources happens to approximate closely the total radiant solar heat that impinges on the window.[1] The various contrary factors apparently offset each other. Therefore the accepted method is to use, for *bare* windows, the values of *I* given in Table 92.

The shading of windows is obviously an economical measure. Inside shades are of some value, but outside awnings or outside Venetian blinds are much more effective. Table 95 shows the amounts of heat entering, with different methods of shading, as compared with a bare window. These percentages are to be applied to the values of *I* in Table 92, and the results represent the total heat gain H_R for the glass. These values are not subject to the addition of conducted heat H_t as in the case of walls.

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[1] Ref. 1, p. 513.

For windows not exposed to the sun the heat gain may be due either to reflected sky radiation or conduction. The latter is easily calculated by the method outlined in Chap. II, using, of course, the appropriate inside and outside temperatures.

When calculating the total heat gain through windows it is sufficiently accurate to use the following procedure:

Consider the total heat gain H as that resulting from sun radiation H_R and neglect the conducted heat H_t. This method is satisfactory except in those cases where the conducted heat gain H_t is greater than either the solar or sky radiation H_R. Under such conditions the sun radiation H_R may be neglected and the total heat gain H considered as resulting only from the conducted heat H_t.

TABLE 95.—HEAT ENTERING WITH VARIOUS METHODS OF SHADING, IN PER CENT OF THAT THROUGH A BARE WINDOW[1]

Bare window	100
Outside Venetian blind	22
Awning	28
Inside shade, fully drawn	45
Inside Venetian blind	58

[1] HOUGHTEN, GUTBERLET, and BLACKSHAW, "Studies of Solar Radiation through Bare and Shaded Windows," *Trans. A.S.H. & V.E.*, 1934.

To illustrate the importance of the heat gain through windows, Fig. 296 shows the results of a test conducted in four rooms on different exposures of a large office building. The cooling load was largely due to the solar radiation through the windows, and the tests give a comparison between the actual and calculated cooling loads. The calculated cooling load is based on the assumption that nearly all of the solar radiation striking the window entered into the cooling load.

354. Lighting and Other Sources.—The gross amount of heat given off by artificial lighting can be easily calculated from the wattage of the lamps by using the factor of 3413 B.t.u. per kw.-hr. It has been demonstrated,[1] however, that not all of this heat must necessarily be considered as part of the cooling load. If the lighting units are high and out of the path of the cooling air currents, the heat from them will not be absorbed but will increase the temperature of the air near the ceiling. In a central-fan system with inlet grilles discharging horizontally near the

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[1] Ref. 2, p. 512.

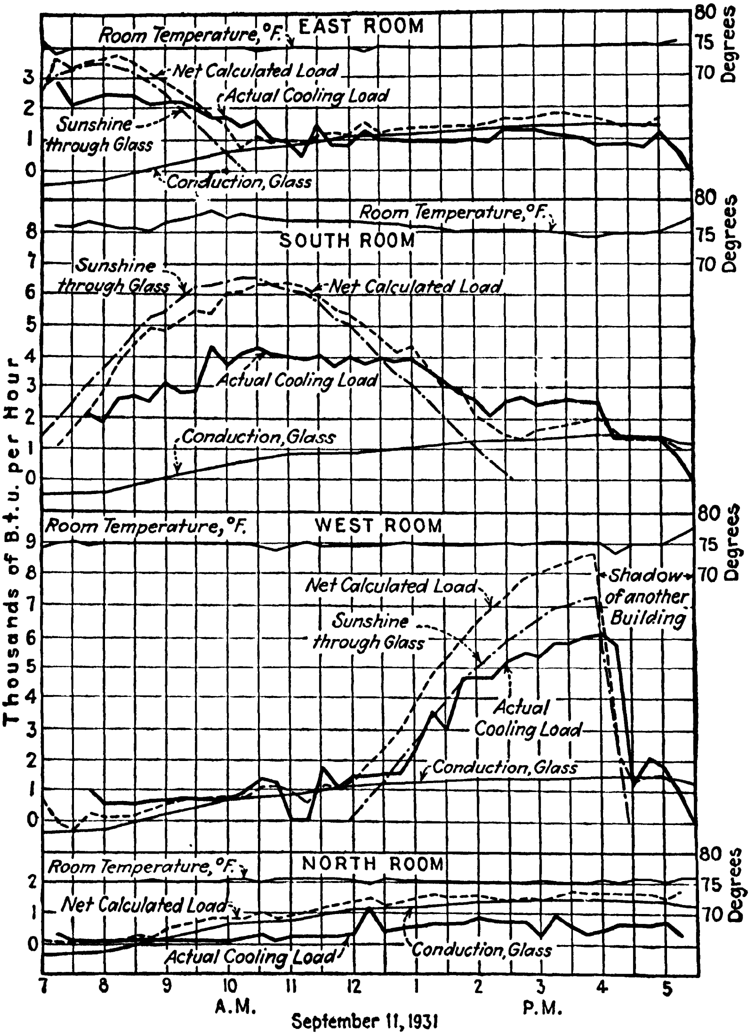

FIG. 296.—Calculated and actual cooling loads in identical rooms on various sides of a building.[1]

ceiling, the full equivalent of the lighting energy should be assumed. The wattage of lighting systems in various kinds of buildings is given in Table 96.

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[1] From WALKER, SANFORD and WELLS, "Field Studies of Office Building Cooling," *Trans.*, *A.S.H. & V.E.*, 1932.

TABLE 96.—CONSUMPTION OF ELECTRICITY FOR LIGHTING

Kind of Building	Electrical Consumption, Watts per Sq. Ft of Floor Area
Office	5
Department store	3 to 5

The question as to what portion of the lamps will be in use during the cooling period requires a study of the room in question, the amount of natural light, and the habits of the occupants.

The heat from motors and machinery can be directly calculated into equivalent heat by the relationship given above, but here again some judgment should be used.

355. Heat and Moisture from Appliances.—In restaurants and in industrial plants the sensible and latent heat from processes may be an important factor. They must be calculated separately. With an electric cooker, for example, the watts input represents the total of the sensible and latent heat, and the latter is the equivalent of the water evaporated. Table 97 gives figures for the more common appliances.

TABLE 97.—HEAT GAIN FROM APPLIANCES

	Sensible heat, B.t.u. per hr.	Latent heat, B.t.u. per hr.
Steam tables, per sq. ft. of top surface:		
Electrically heated	150	50
Gas heated	400	900
Steam heated	600	1,600
Coffee urns, per gal. rated capacity:		
Electrically heated	600	600
Gas heated	700	700
Steam heated	600	600
1 cu. ft. natural gas per hr. generates	900	100
1 cu. ft. artificial gas per hr. generates	540	60
1 cu. ft. producer gas per hr. generates	135	15
Heat liberated by food per hr. per person in a restaurant	30	30

356. Infiltration and Ventilation.—When no outside air is introduced, it is necessary to consider only the natural infiltration through leakage. For winter conditions this was discussed in Chap. II. In summer the wind velocities on warm days are usually low, and the infiltration may safely be assumed at one air renewal per hour.

Usually, however, the system will be designed to introduce a definite quantity of outside air. The heat to be absorbed by the cooling and dehumidifying apparatus is the difference between the heat content, per pound, of the outside air and the heat content of the air delivered by the apparatus. The total heat of the air and the moisture content for the two conditions can be readily determined from the psychrometric chart and the differences obtained.

The amount of outdoor air introduced for ventilation is usually based upon the number of occupants. The necessity for introducing outdoor air in summer is purely a matter of restricting the building up of odors. In offices, theaters, and the like, 10 to 20 c.f.m. per person is ordinarily assumed. Twenty cubic feet per minute gives ample ventilation, and 10 is regarded as the very minimum. Where there is much smoking the larger figure is needed.

357. Occupants.—The presence of human beings has an effect, sometimes very important, on both the cooling and the dehumidifying load. In a theater the load from the occupants is the major part of the total cooling load.

In Fig. 202 (Chap. XIV) curves are given of the sensible heat loss from the human body under various conditions. For men seated at rest, the sensible heat loss for a temperature of 80° is 220 B.t.u. per hr. per person. The total heat emitted is approximately 400 B.t.u. per hr. per person, but part of this is in the form of latent heat in the moisture given off.

The moisture given off by the occupants is shown in Fig. 203 (Chap. XIV). For men seated and at rest at 80° it amounts to 1,200 gr. per hr. per person, which is equivalent to 175 B.t.u.

The moisture loss increases greatly with mild activity—more so than the heat loss—as shown by the difference between curves *C* and *D* (Fig. 203). Therefore the designer should carefully ascertain the degree of activity to be expected, as it may exert an important effect upon the total load.

358. Summary.—By means of the methods given in the foregoing pages the various components of the cooling and dehumidifying load should be determined hour by hour and for each room or each typical room of the building. The total load for the building is determined by summing up hour-by-hour loads of the various rooms and adding the load due to the outdoor air introduced.

Note that the heat equivalent of the power input to the fan is a component; and if the ducts extend through uncooled spaces, this may be an additional item of load.

Example.—Calculate the cooling and dehumidifying load for a typical case with the following data given:

Office room, 150 by 60 by 12 ft. high.

Construction, 8-in. brick, lath, and plaster. $U = 0.30$.

Outer net wall area, 1,344 sq. ft. facing west, 624 sq. ft. north.

Glass area, 552 sq. ft., of which 456 sq. ft. is due-west exposure and 96 sq. ft. north exposure.

Occupants, 100.

Lighting, 27 kw.

No awnings, but window shades.

Ventilation, 15 c.f.m. per person.

Indoor and outdoor conditions:

Sun time	Indoors, deg.		Outdoors, deg.	
	Dry-bulb	Wet-bulb	Dry-bulb	Wet-bulb
10:00 A.M.	75	63	80	66
11:00	75	63	84	68
12:00 M.	75	63	88	70
1:00 P.M.	75	63	92	72
2:00	76	63	94	74
3:00	78	64	97	76
4:00	80	65	95	74
5:00	80	65	90	72
6:00	75	63	85	70
7:00	75	63	80	66

Wall transmission:

North wall:

$$H_t = 624 \times 0.30(t_o - t).$$

West wall:

$$H_t = 1{,}344 \times 0.30(t_o - t).$$
$$H_R = 0.23 \times 1{,}344 \times 0.30 \times 0.7 \times I = 64.9I.$$

Assume time lag to be 3 hr. (from Table 94 by inspection).

H_t for north and west walls combined $= 590.4(t_o - t)$.

Sun time	$(t_o - t)$, deg.	H_t, B.t.u.	I, B.t.u.	H_R with 3-hr. lag	Total wall transmission, B.t.u. per hr
10:00 A.M.	5	2,960	23.5	......	2,960
11:00	9	5,310	25.5	......	5,310
12:00 M.	13	7,680	26	......	7,680
1:00 P.M.	17	10,050	94	1,520	11,570
2:00	18	10,650	152	1,650	12,300
3:00	19	11,250	195	1,690	12,940
4:00	15	8,860	211	6,100	14,960
5:00	10	5,920	162	9,870	15,790
6:00	10	5,920	56	12,670	18,590
7:00	5	2,960	6	13,690	16,650

Glass transmission:

North side:

$$H_g = 96 \times 1.13(t_0 - t) = 108.5(t_0 - t) \text{ B.t.u. per hr.}$$

West side (in sun):

$$H_g = 0.45^* \times 456 \times I = 206I$$

Sun time	$(t_o - t)$, deg.	H_g north, B.t.u.	I	Sun heat through glass, B.t.u.	Total heat through glass, B.t.u.
10:00 A.M.	5	540	23.5	4,840	5,380
11:00	9	980	25.5	5,250	6,230
12:00 M.	13	1,410	26	5,360	6,770
1:00 P.M.	17	1,840	94	19,350	21,190
2:00	18	1,950	152	31,300	33,250
3:00	19	2,060	195	40,200	42,260
4:00	15	1,630	211	43,400	45,030
5:00	10	1,090	162	33,400	34,490
6:00	10	1,090	56	11,520	12,610
7:00	5	540	6	1,230	1,770

Ventilation:

15 c.f.m. × 100 people × 60 min. = 90,000 c.f.h.

Assume 13.84 cu. ft. per lb.

Weight of air per hr. = 90,000 ÷ 13.84 = 6,510 lb. per hr.

The total heat removed is the difference between the Carrier total heats for the outdoor and indoor conditions for this amount of air. The latent heat removed is that corresponding to the difference between the inside and outside moisture contents.

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* Inside shades, Table 95.

Actually the outdoor air that is introduced is cooled and dehumidified to a condition of lower total heat than that of the room air. From a heat-balance viewpoint, however, the heat chargeable to the ventilation air is obviously that equivalent to the difference between the total heat of the indoor and outdoor air.

Sun time	Carrier total heat per lb. outside, B.t.u.	Carrier total heat per lb. inside, B.t.u.	Moisture outside, gr. per lb.	Moisture inside, gr. per lb.	Carrier total heat diff., B.t.u. per lb.	Moisture diff., gr. per lb.	Latent heat[1] diff., B.t.u. per lb.	Sensible heat diff., B.t.u. per lb.
10:00 A.M.	30.3	28.1	73	67	2.2	6	0.91	1.29
11:00	31.9	28.1	77	67	3.8	10	1.51	2.29
12:00 M.	33.4	28.1	81	67	5.3	14	2.12	3.18
1:00 P.M.	35.1	28.1	86	67	7.0	19	2.87	4.13
2:00	36.8	28.1	94	65	8.7	29	4.38	4.32
3:00	38.7	28.8	102	67	9.9	35	5.30	4.60
4:00	36.8	29.6	92	68	7.2	24	3.62	3.58
5:00	35.1	29.6	89	68	5.5	21	3.17	2.33
6:00	33.4	28.1	86	67	5.3	19	2.87	2.43
7:00	30.3	28.1	73	67	2.2	6	0.91	1.29

[1] Latent heat = (gr. per lb./7,000) × 1,059.

The values used in this illustrative example have been read from the psychrometric chart, which is permissible unless considerable accuracy is desired, in which case Table 66 should be used.

Occupants:

There are 100 occupants.
Sensible heat = 220* × 100 = 22,000 B.t.u. per hr.
Latent heat = 180† × 100 = 18,000 B.t.u. per hr.

Lighting

Assume that of the 27 kw. of lighting, one-third, or 9 kw., will be in use during the time of maximum cooling and that the method of introduction of the air is such that all of this heat must be absorbed.

$$H = 9 \times 3{,}413 = 30{,}700 \text{ B.t.u. per hr.}$$

Summary:

The next step is to tabulate the hourly loads as in Table 98.

The total maximum load, except for a few minor items, is 190,300 B.t.u. per hr., or 15.86 tons of refrigeration, and it occurs at 3:00 P.M. sun time.

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* Fig. 202, Chap. XIV.

† Fig. 203, Chap. XIV.

TABLE 98.—SUMMARY OF HOURLY LOADS, IN B.T.U.

	10:00	11:00	12:00	1:00	2:00	3:00	4:00	5:00	6:00	7:00
Walls	2,960	5,310	7,680	11,570	12,300	12,940	14,960	15,790	18,590	16,650
Glass	5,380	6,230	6,770	21,190	33,250	42,260	45,030	34,490	12,610	1,770
Ventilation (sensible)	8,400	14,900	20,700	26,900	28,100	29,900	23,300	15,200	15,800	8,400
Occupants (sensible)	22,000	22,000	22,000	22,000	22,000	22,000	22,000	22,000	22,000	22,000
Lights	30,700	30,700	30,700	30,700	30,700	30,700	30,700	30,700	30,700	30,700
Total of sensible heat	69,440	79,140	87,850	112,360	126,350	137,800	135,990	118,180	99,700	79,520
Ventilation (latent)	5,930	9,830	13,800	18,700	28,500	34,500	23,600	20,700	18,700	5,930
Occupants (latent)	18,000	18,000	18,000	18,000	18,000	18,000	18,000	18,000	18,000	18,000
Total of latent heat	23,930	27,830	31,800	36,700	46,500	52,500	41,600	38,700	36,700	23,930
Total load[1]	93,370	106,970	119,650	149,060	172,850	190,300	177,590	156,880	136,400	103,450

[1] Except for duct losses, heat from fan power, etc.

359. Cooling and Dehumidifying Relationships.—One feature of cooling by refrigeration, the interrelation of the cooling and dehumidifying effects, is both advantageous and troublesome. The method of dehumidifying by means of refrigeration consists of cooling the air below its dew-point temperature, and the degree of dehumidification depends upon the new dew point to which it is brought.

The conditions produced by a spray washer are slightly different from those produced by a coil. With a washer, if it is equipped with two or more banks of sprays, the air is cooled to saturation; and regardless of the leaving-air temperature, the point representing the leaving condition on the psychrometric chart always lies on the saturation line. With a coil the air is cooled and dehumidified; but unless a very deep coil is used, the leaving air is not at saturation.

Considering first the conditions with a spray washer, and referring to Fig. 297, if air at condition *A*, assumed to be 85° dry bulb and 70° wet bulb, is cooled to its dew point *B*, which is 63°, it will be just saturated, and no moisture will have been removed. If it is further cooled successively to points *C*, *D*, *E*, etc., corresponding to temperatures 60°, 55°, 50°, etc., increasing quantities of moisture will be condensed.

In Tables 99, columns 2, 3, and 4 give conditions of the air at the various points in Fig. 297. In columns 5 and 6 are given, respectively, the total heat and sensible heat removed in passing from the initial point *A*.[1]

Column 7 gives the **load ratio** of the sensible heat removal, in B.t.u. per pound, to the total heat removal, in B.t.u. per pound.

Therefore if air at the conditions of point *A* is conditioned and brought down to any of the points *C*, *D*, *E*, etc., the ratio of sensible to total heat removal will be as shown in column 7. This ratio may be changed within certain limits, to fit the conditions imposed by the cooling and dehumidifying requirements of the building.

This will be perhaps more thoroughly understood by reference to Fig. 297. If air is cooled and dehumidified from condition *A* to *F*, the moisture removal is proportional to the distance *LH* on the chart, and the sensible heat removal is proportional to the distance *SH*. The ratio *SH* to *TH* (*LH* plus *SH*) does not vary according to any simple law, and in this form of psychrometric chart the desired dew point can be determined only by trial.

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[1] Computed from the formula $h = 0.24(85 - t)$, in which 0.24 is the specific heat of air and t = the temperature at *B*, *C*, *D*, etc.

For example, assume that the total moisture to be removed per hour is 249,900 gr., which is equivalent to a latent heat load of (249,900/7000) × 1059 = 37,800 B.t.u. per hr. The sensible heat load is 155,651 B.t.u. per hr., and the total is 37,800 + 155,651 = 193,451 B.t.u. per hr. The resulting *load ratio* is then 155,651/193,451 or 0.804. Assuming temporarily that the air entering the washer has the conditions of point *A*, then the required dew point will be slightly below 60° (for which the load ratio is 0.83) or, say, 59°.

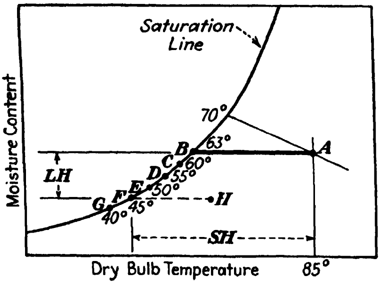

FIG. 297.

The sensible heat between 85 and 59° is 0.24(85 − 59) = 6.24 B.t.u. per lb. To calculate the amount of air to be dehumidified, first calculate the density of air at 85° dry bulb and 70° wet bulb (from the tables of the properties of air) to be 0.0723 lb. per cu. ft. Then

TABLE 99.—CONDITIONS AT POINTS IN FIG. 297

Points	Temperatures, °F.		Moisture content, gr. per lb. of air	Carrier total heat, B.t.u. per lb.	Total heat removed B.t.u. per lb.	Sensible heat removed, B.t.u. per lb.	Load ratio Col. 6 / Col. 5
	d.b.	w.b.					
1	2		3	4	5	6	7
A	85	70	86	33.4			
B	63	63	86	28.1	5.3	5.3	1.00
C	60	60	77	26.1	7.3	6.00	.83
D	55	55	64	23.0	10.4	7.20	.69
E	50	50	53	20.1	13.3	8.40	.63
F	45	45	44	17.5	15.9	9.60	.60
G	40	40	36	15.2	18.2	10.80	.59

$$\text{C.f.m.} = \frac{155{,}651}{60 \times 6.24 \times 0.0723} = 5{,}730$$

which is the quantity to be passed through the coils or washer at the time when the calculated maximum load conditions prevail.

360. Performance of a Coil.—The foregoing analysis applies to a spray washer. With a coil the exit condition is not at saturation but at some relative humidity depending upon the depth of coil used. If the inlet and exit conditions are located on a nonlogarithmic chart such as Fig. 194 (Chap. XIII), the angle that a straight line connecting the two points makes with the horizontal is a direct measure of the relative amounts of latent to sensible heat removal.

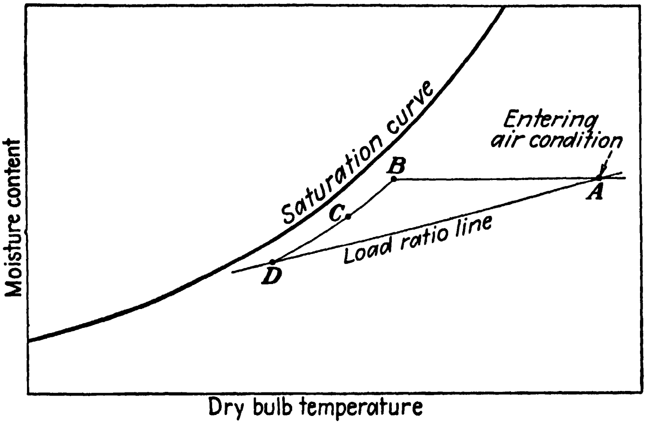

FIG. 298.—Diagram showing load-ratio line.

Suppose air enters a cooling coil at conditions corresponding to point A in Fig. 298. As long as the coil surface temperature is above the air dew point, dry cooling takes place until B is reached, which is that point where the coil surface temperature is exactly equal to the air dew point. If the coil temperature is reduced below the dew point, dehumidification takes place, and the air will have a condition somewhere along the line B, C, D, which is a curve at a *constant horizontal distance* from the saturation curve. Hence the minimum dry bulb for no dehumidification (point B) and the exit conditions for any load ratio should have approximately the same relative humidity.

A line through AC or AD represents the commonly known load ratio line referred to in Art. 359. It should be understood that line $ABCD$ is not intended to indicate anything about the air condition as it passes through the coil from row to row. It simply represents the possible location of the condition of the air as it *leaves* a coil operating with decreasing surface temperature.

The foregoing comparison emphasizes the advantage of the washer over the coil, in accuracy of control. With the washer the exact dew point of the leaving air can be fixed by controlling the spray-water temperature, and the leaving conditions will not vary because of changes in the entering-air conditions. Where close control is necessary, as in industrial air conditioning, the washer is therefore superior. Its disadvantages are the space occupied and the maintenance required.

It has been shown that both the sensible heat and the moisture loads could be taken care of, within limits, by choosing the proper dew point and by passing just the proper quantity of air through the coil or washer. To do this with a fluctuating *load ratio* and under varying load conditions would call for frequent readjustments that are not feasible in practice. Therefore, a dew point is chosen which is satisfactory for average conditions and which permits the humidity in the rooms to vary. For comfort air conditioning this is permissible because of the rather wide range of comfortable conditions. Also it has been shown in Art. 235, Chap. XIV, that the effect of moderate changes in humidity upon the feeling of warmth is not great.

361. Reheating.—Suppose, however, that the ratio of sensible to total heat removal were much lower than any of the figures in column 7 of Table 99. This would be the case in a theater or perhaps a restaurant. Then to remove the required amount of moisture would require the removal of more heat. If necessary, the excess heat removed could be restored by reheating the air after dehumidification. This is sometimes done, particularly in industrial work if both temperature and humidity must be accurately controlled, but ordinarily the practical solution is to control the dry-bulb temperature only, permitting the humidity to fluctuate.

On the psychrometric chart (Fig. 297) the process of reheating is indicated by the line *FH*. The dew point *F* would be selected to give the proper total moisture removal for the quantity of air to be passed through the system, and the point *H* would be so chosen that the sensible heat difference between *H* and *A* would be equivalent, when multiplied by the air quantity, to the required amount of sensible heat to be removed. Reheating is obviously uneconomical because it requires removal, by refrigeration, of heat that is deliberately added to the system.

362. By-passing and Other Arrangements.—To keep down the size of the apparatus, it is advisable to condition only a moderate quantity of air. It has been shown that the *load ratio* controls the quantity to be passed through the coil or washer and that the exit temperature required depends also on this ratio. But because of drafts and uneven temperatures it is undesirable to introduce extremely cold air to the rooms. One obvious way of avoiding this difficulty is to mix the cooled air with some of the room air before introducing it to the room. Figure 299 shows such an arrangement which is covered by patents and is commonly known as the "by-pass" method. If the moisture load is relatively large, a lower dew point is carried, and vice versa.

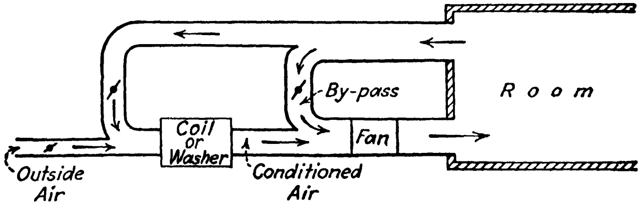

Fig. 299.—Diagrammatic arrangement of by-pass method.

As the total load increases, a larger volume of air is passed through the coil or washer, and the proportion may be automatically controlled. The advantages of this method in addition to that mentioned are that it permits a constant total quantity of air to be circulated (a varying proportion being by-passed) and lends itself readily to control for varying loads. It is undersirable to vary the air volume in a large duct system, because it may seriously affect the distribution.

The lowest temperature at which it is permissible to introduce air into a room will depend upon the ceiling height and the direction of the entering streams of air. Of course it is desirable not to have a large difference between the entering temperature and the room temperature. The smaller this temperature difference, the greater the quantity of air that must be circulated to pick up the sensible heat load of the room. If the method of introduction of the air is carefully planned, a temperature difference of 20° is permissible, but this is regarded as the maximum.

363. Calculation of Air Quantity.—It is desirable to reduce the quantity of air to be circulated to the rooms to the lowest practicable quantity, in order to keep down the sizes of ducts, fans, etc. The quantity can be readily calculated from the sensible heat gain and the temperature rise.

The air quantity for the example of Art. 358 may be calculated in the following manner. It is, of course, only the heat load of the room itself that must be taken into account—not the heat in the outdoor air drawn in by the fan. For these items the maximum occurs at 4 P.M. and the items are (Table 98)

	B.t.u. per Hr.
Walls	14,960
Glass	45,030
Occupants	22,000
Lights	30,700
Total	112,690

The room temperature is to be 80°, and it may be assumed that the air will be introduced at 65° dry bulb.

The quantity of air required will be

$$\frac{112,690}{60 \times 0.241(80 - 65)} = 519 \text{ lb. per min.}$$

The actual volume of air at 65° will be 519 × 13.5 = 7,000 c.f.m.

The room volume is 150 by 60 by 12 ft. = 108,000 cu. ft.

The number of air renewals per hour = 7,000 × 60/108,000 = 3.9.

This is a conservative figure. Under conditions of a high heat load per cubic foot of room volume, the air quantity may be equivalent to eight or ten air renewals per hour, and it is difficult to circulate that much air without causing drafts. Quantities up to six or seven air renewals per hour can usually be introduced without much trouble.

Because of the heat gain in the ducts, which usually pass through warm spaces, the temperature of the air leaving the coil or spray chamber must be appreciably lower than the temperature entering the rooms. The method of calculating heat loss or gain in ducts was explained in Art. 309, Chap. XVII.

364. Control for Fan Systems.—Thermostatic control is absolutely necessary on most types of fan systems. Hot-blast systems in factories are almost the only exceptions. Many different arrangements of the thermostats and valves may be used, depending upon the results desired. Figure 300 shows a method of applying thermostatic control to a ventilating system.

The tempering unit A is controlled solely by the fresh-air temperature, the thermostat being set for 35°, so that this unit will be full of steam at temperatures below that point. The air washer is equipped with a water heater controlled by its own thermostat. Reheating unit B is controlled by a thermostat located in the tempered-air chamber and adjusted so that the temperature is approximately 68°. Units C and D are hand-controlled, according to the heating requirements, the temperature in the hot-air chamber being carried higher in colder weather.

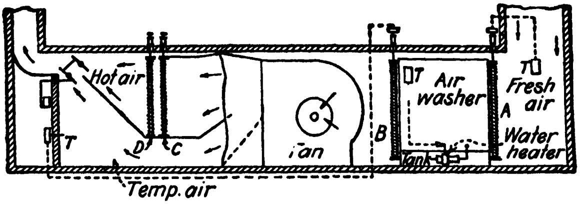

FIG. 300.—Thermostatic control applied to a fan system.

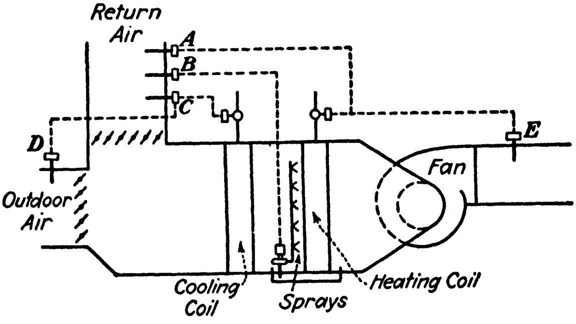

FIG. 301.—Control for winter and summer operation.

The mixing dampers, of which there are a pair for each room in the building, are controlled by thermostats located in the respective rooms.

Control for systems designed for summer cooling may consist of a simple valve regulating the chilled water to the cooling coil and controlled by a room thermostat, or it may be quite complete and complex. There are so many possible combinations of elements and so many ingenious devices that it is impracticable here to describe many. The catalogues of manufacturers are helpful in a study of detailed arrangements.

A rather simple control system for both winter and summer operation is shown in Fig. 301. This arrangement is suitable only when the system supplies a single room or a group or interconnected rooms. It does not provide for separate room control.

The steam supply to the heating coil is regulated by a modulating steam valve controlled by the thermostat *A* placed in the return duct. The return air, of course, is at very nearly room temperature. The thermostat *E* in the supply duct prevents the supply-air temperature from falling to an uncomfortably low level.

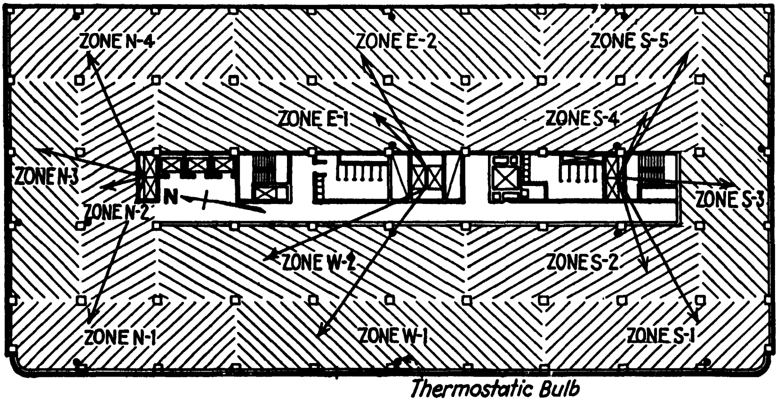

FIG. 302.—Zoning arrangement in an office building.[1]

Humidity in winter is controlled by the humidistat *B*, which starts and stops the spray pump. The chilled water to the cooling coil is controlled by the thermostat *C* and the modulating valve in the chilled-water supply pipe. Thermostat *D*, affected by the outdoor air temperature, modifies the effect of the thermostat *C* so that the indoor temperature rises as the outdoor temperature rises, to give the relationship of the two temperatures as described in Art. 238 and shown in Table 67, Chap. XIV. This method of differential control, which may be accomplished by several different mechanical or electrical devices, is in rather common use.

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[1] JAMES N. LIVERMORE, "Unusual Air Conditioning Problems Met in Designing New Glass Block Building," *Heating, Piping and Air Conditioning*, April, 1938.

365. Zoning.—The zoning of a cooled building to take care of the varying heat gain by the different rooms or exposures as the sun travels around the building is quite important, especially if the sun effect through the windows is an important part of the cooling load. Figure 302 shows a large office building divided into several zones per floor, as shown by the crosshatching. The arrows represent diagrammatically the air ducts supplying the various zones from the vertical trunk ducts. A separate thermostat is required for each zone. This method of control is satisfactory only if there are no subdivisions of the space comprising each zone. It is not satisfactory to control more than one room from a single thermostat.

366. Selection of Cooling Coils.—The transfer of heat between the cooling medium of a coil and the air stream is governed by several factors. They are air velocity, temperature difference, design and surface arrangement of coil, and velocity and nature of fluid inside tubes. Because of the many complex and indeterminate variables involved in predicting the performance of cooling coils, numerous empirical systems have been developed by manufacturers for selecting this type of equipment over a *limited range* of operating conditions.

To simplify the procedure of selecting coils for direct expansion, a method[1] utilizing three simple formulas and a psychrometric chart has been devised. From these formulas the size and depth of coil can be computed, the actual condition of the air as it leaves the coil may be established, and the refrigerant temperature can be determined. With these results it is possible to select a manufacturer's coil and predict its performance at almost any point over its *entire operating range.*

The relationship between number of rows of coil depth, leaving-air condition, and outside-surface coefficient is expressed by the following equation

$$\frac{h_o A_s N}{0.24W} = \log_e \left(\frac{t_1 - dp_1}{t_2 - dp_2}\right) \tag{7}$$

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[1] Tuve and Seigel, "Performance of Surface-coil Dehumidifiers for Comfort Air Conditioning," *Trans. A.S.H. & V.E.*, 1938; and Seigel, "Air Cooling Coil Problems and Their Solutions," *Jour. A.S.H. & V.E.*, February, 1945.

where h_o = outside-surface coefficient (air side), B.t.u. per hr. per sq. ft. per deg. F. For many conventional types of coils $h_o = 1.1\ (dV)^{0.6}$. where d = density of air, lb. per cu. ft., and V = air velocity, ft. per min.
A_s = outside surface area, sq. ft. per sq. ft. of coil face area per row of coil depth.
N = number of rows of coil depth.
W = weight of air-vapor mixture, lb. per hr. per sq. ft. coil face area.
t_1 = dry-bulb temperature air entering coil, deg. F.
t_2 = dry-bulb temperature air leaving coil, deg. F.
dp_1 = dew-point temperature air entering coil, deg. F.
dp_2 = dew-point temperature air leaving coil, deg. F.

The coil-surface temperature may be determined from Eq. (8) and the refrigerant temperature by solving Eq. (9).

$$t_s = \frac{\left(\frac{t_1 - dp_1}{t_2 - dp_2}\right) t_2 - t_1}{\left(\frac{t_1 - dp_1}{t_2 - dp_2} - 1\right)} \tag{8}$$

where t_s = coil-surface temperature, deg. F.

$$t_r = t_s - \frac{H_t R}{N A_s h_r} \tag{9}$$

where t_r = refrigerant temperature, deg. F.
H_t = total coil load, B.t.u. per hr. per sq. ft. coil face area.
h_r = inside-surface coefficient (refrigerant side), B.t.u. per sq. ft. per deg. F. A value of $h_r = 325$ is satisfactory for Freon.
R = ratio air side to refrigerant-side surface area.

The following example will illustrate the procedure when the entering-air condition together with load ratio are known and it is desired to establish the actual leaving-air condition, depth of coil, and refrigerant temperature.

Example.—It is desired to select a coil to maintain room conditions of 80° dry-bulb and 67° wet-bulb temperature when the calculated load has been determined as 60,000 B.t.u. per hour and the *load ratio* is 0.65. The coil is to cool and dehumidify 1,450 c.f.m. with a net face-area velocity through the coil of 600 ft. per min.

Limited duct space makes it necessary to circulate this minimum quantity of air with a resulting differential of about 25° between room-temperature and air-entering conditions. This is in excess of the permissible temperature-differential limit of 20° mentioned in Art. 362, but with an injector or high-induction system of air distribution (Art. 315) it is possible to avoid drafts where it is found necessary to circulate low-temperature air.

Assume coil under consideration has 22 sq. ft. external surface, including fins and tubes, per sq. ft. face area per row of coil depth. Also this coil has a ratio of air-side to refrigerant-side surface area of 15.

Determine the coil depth, face area, refrigerant temperature, and the actual air conditions leaving the coil.

With a load ratio equal to 0.65, the sensible heat load may be calculated as $0.65 \times 60{,}000 = 39{,}000$ B.t.u. per hr., and by difference the latent heat load amounts to 21,000 B.t.u. per hr. The volume of room air mixture from the psychrometric chart (Figs. 194 and 303) is 13.84 cu. ft. per lb., or the density is $1/13.84 = 0.0723$ lb. per cu. ft. Also, the dew point dp_1 of the entering-air conditions is found on the chart to be 60.3°.

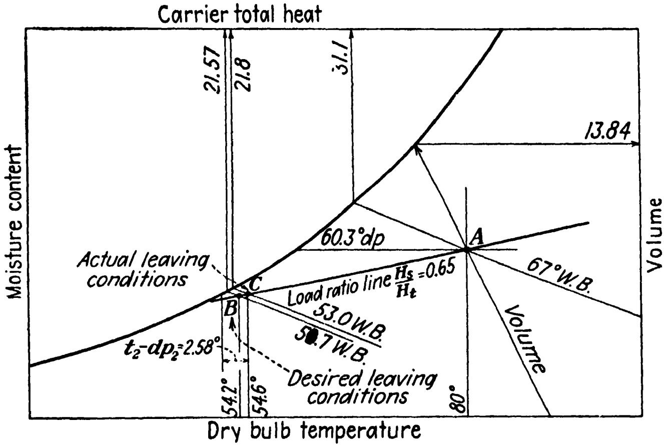

FIG. 303.—Cooling-coil selection diagram.

Next check the amount of sensible and total heat that must be extracted per pound of air circulated in order that the actual leaving conditions may be estimated.

$$\frac{H_s}{60Qd} = \frac{39{,}000}{60 \times 1450 \times 0.0723} = 6.2 \text{ B.t.u. per lb. air}$$

$$H_s = C_pW\,(t_1 - t_2)$$

$$6.2 = 0.24 \times 1(t_1 - t_2)$$

$$(t_1 - t_2) = \frac{6.2}{0.24} = 25.8°$$

With $t_1 = 80°$, then $(80 - 25.8) = t_2 = 54.2°$

$$\frac{H_t}{60Qd} = \frac{60{,}000}{60 \times 1450 \times 0.0723} = 9.53 \text{ B.t.u. per lb. air}$$

From the psychrometric chart it will be found that the Carrier total heat corresponding to entering-air conditions A of 67° wet bulb is 31.1 B.t.u. per lb. air. Subtracting, $31.1 - 9.53 = 21.57$ B.t.u. per lb. is the Carrier total heat of the leaving-air conditions. This corresponds to a wet-bulb temperature $t'_2 = 52.7°$ which intersects the dry-bulb temperature $t_2 = 54.2°$, at a dew point of $dp_2 = 51.8°$, thus establishing the *desired* air conditions B leaving the coil. Connecting points A and B in Fig. 303 give the load ratio line for this example.

Before substituting the proper values in Eq. (7) to determine the coil depth, it is necessary to calculate the weight of air circulated, W. The coil net face area is obtained by dividing the air quantity by the velocity or $1450/600 = 2.42$ sq. ft. Then

$$W = \frac{60 \times 1450 \times 0.0723}{2.42} = 2600 \text{ lb. per hr. per sq. ft. coil face area}$$

Also, it is necessary to calculate the outside surface coefficient $h_o = 1.1(0.0723 \times 600)^{0.6} = 9.6$ B.t.u. per hr. per sq. ft. per deg. F.

Substituting in Eq. (7)

$$\frac{9.6 \times 22 \times N}{0.24 \times 2600} = \log_e \left(\frac{80 - 60.3}{54.2 - 51.8}\right)$$
$$N = 6.21$$

This establishes the whole number of coil rows that should be used as 6 or 7, and it is now possible to determine the *actual* location of the leaving conditions from Eq. (7) by solving for the actual value $(t_2 - dp_2)$ for a 6-row coil.

$$\frac{9.6 \times 22 \times 6}{0.24 \times 2600} = \log_e \left(\frac{80 - 60.3}{t_2 - dp_2}\right)$$
$$\text{antilog}_e\, 2.03 = 7.61 = \left(\frac{80 - 60.3}{t_2 - dp_2}\right) = \left(\frac{t_1 - dp_1}{t_2 - dp_2}\right)$$

and

$$(t_2 - dp_2) = 2.58°$$

Next, locate point C on the load-ratio line at a horizontal distance of 2.58° dry bulb from the saturation curve. This is the only part of the procedure that requires a graphical determination. The actual leaving conditions are thereby established as $t_2 = 54.6°$; $t'_2 = 53.0°$; $dp_2 = 52.0°$; and Carrier total heat = 21.8 B.t.u. per lb.

The coil-surface temperature may now be found from Eq. (8).

$$t_s = \frac{(7.61 \times 54.6) - 80}{(7.61 - 1)} = 50.7°$$

Total coil load may be calculated from the total heat difference between points A and C.

$H_t = W(h_a - h_c) = 2600(31.1 - 21.8) = 24{,}200$ B.t.u. per hr. per sq. ft. of face area. Equivalent tons of refrigeration 24,200 × 2.42 sq. ft. face area = 58,500 B.t.u. per hr. or 4.875 tons of refrigeration.

Refrigerant temperature can be calculated from Eq. (9)

$$t_r = 50.7 - \frac{24{,}200 \times 15}{6 \times 22 \times 325} = 42.23°$$

Summarizing, the coil characteristics are as follows: depth of coil = 6 rows; face area = 2.42 sq. ft.; refrigerant temperature = 42.23°; leaving conditions, dry bulb = 54.6° and wet bulb = 53.0°. Using these data, a suitable coil can be selected.

In direct-expansion systems the refrigerant temperature is that corresponding to the suction pressure of the refrigerant. The refrigerant is automatically maintained at approximately the same temperature in all sections of the coil, except that a moderate amount of superheating (about 10°) is desirable to avoid the possibility of the liquid refrigerant's entering the suction of the compressor, with consequent danger, in a reciprocating machine, of damage to the compressor.

Where chilled water is used, the condition is somewhat different. Since the water flow through the coils is in series, there is an appreciable temperature rise (10 to 15°). The heat transfer in the unit as a whole depends upon the **mean temperature difference** between the air and the water in the tubes. This may be calculated from the following relationship:

$$\text{M.T.D.} = \frac{(t_1 - t_{w_2}) - (t_2 - t_{w_1})}{\log_e \dfrac{t_1 - t_{w_2}}{t_2 - t_{w_1}}} \tag{10}$$

where M.T.D. = mean temperature difference, deg. F.
t_1 = temperature of entering air.
t_2 = temperature of leaving air.
t_{w_1} = temperature of entering water.
t_{w_2} = temperature of leaving water.

The amount of surface is then determined by the formula

$$A = \frac{H}{U \times \text{M.T.D.}} \tag{11}$$

where A = coil surface, sq. ft.
H = *sensible* heat to be removed, B.t.u. per hr.
M.T.D. = mean temperature difference, deg. F.
U = over-all transmission coefficient, B.t.u. per sq. ft. per hr. per degree difference.

Until a simple and direct method of computation is derived for ascertaining the air-side and water-side surface coefficients for water coils, it is necessary to use the over-all coefficient U and Eq. (10) for selecting such equipment. This latter factor is most difficult to determine, as it depends upon the many variations in coil design and water velocity.

The quantity of water circulated depends upon the total amount of heat removed from the air (sensible heat plus latent heat) and the temperature rise of the water that is to be allowed.

Other items to be calculated are the face area of the units, the air-friction drop, and the water-friction drop. The net face area is usually based on a velocity of about 500 ft. per min. The friction drops are best taken from the manufacturer's data.

Provision must be made for draining off the water of condensation from the coil casing.

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Problems

1. Calculate the sensible cooling load, moisture load, total peak refrigeration, and time of peak load required for the following case:

6,300 sq. ft. net, of southeast wall, $U = 0.27$.

780 sq. ft. of window glass, southeast exposure, not externally shaded but with inside window shades.

Occupants, 227.

Lighting in use, 17 kw.

Motors in use aggregating 15 hp.

Outside conditions:

5:00 to 7:00 A.M.	70° dry bulb, 60° wet bulb
7:00 to 9:00 A.M.	75° dry bulb, 63° wet bulb
9:00 to 11:00 A.M.	80° dry bulb, 66° wet bulb
11:00 to 1:00 P.M.	85° dry bulb, 69° wet bulb
1:00 to 3:00 P.M.	90° dry bulb, 72° wet bulb
3:00 to P.M.	95° dry bulb, 75° wet bulb

Inside conditions, 75° dry bulb, 65° wet bulb.

Outside air, 10 cu. ft. per min. per occupant.

The authors' answer is as follows:

Sensible cooling load	211,300 B.t.u. per hr.
Moisture load	73,800 B.t.u. per hr., or 69.8 lb. per hr.
Total peak refrigeration load	23.7 tons
Time of peak load	3:00 P.M.

The answer will vary, depending on the assumptions made.

2. In the air-conditioning system shown in Fig. 299 the following conditions exist:

Location	Dry-bulb temp.	Wet-bulb temp.	Lb. air per min.
Outside air	95°	75°	300
Entering room	65°	55°	3,000
Leaving room by return duct	85°	65°	2,700

How much air should pass through the by-pass, and how much should go through the washer?

3. A room has a moisture load of 53,000 gr. per min. and a sensible heat load of 10,500 B.t.u. per min. It must be maintained at 75° dry-bulb and 40 per cent relative humidity. This is to be accomplished by a spray-type dehumidifier operated to give a leaving-air temperature of 40°, saturated, followed by reheating.

(*a*) How much air must be put through the dehumidifier? (*b*) How much reheat must be added per minute? (*c*) What is the dry-bulb and wet-bulb temperature of the air entering the room?

4. The air entering a coil has a dry-bulb temperature of 83° and a dew point of 70°. The average temperature of the coil surface is 50°. The air leaves the coil at a dry-bulb temperature of 59°. What is the load ratio?

CHAPTER XX

THE DESIGN OF AN AIR-CONDITIONING SYSTEM

367. Introductory.—The purpose of this chapter is to illustrate, step by step, the design of a hypothetical heating and air-conditioning system. It is intended to illustrate the sort of conditions that the engineer encounters and has purposely been selected because it has some unusual features. It will probably be noticed that some of the assumptions made are not based on any rules given in previous chapters. In practice the exercise of judgment is often necessary, because there are no rules to apply. The general principles employed, however, are those outlined in the preceding chapters.

In the thermal calculations the properties of air are taken largely from Table 66, Chap. XIII, rather than from the psychrometric chart. This is done for greater accuracy. In using Table 66 it is necessary merely to remember that the enthalpy or total heat of air at a given wet-bulb temperature is the same as the total heat of *saturated* air at the same temperature. Also, the latent heat is obtained by subtracting from the total heat the heat content per pound of dry air at the stated dry-bulb temperature.

Figures 304 and 305 show a plan of the first floor and elevations of a small three-story office building in Detroit. The owner plans to rearrange the partitions, revamp the heating system, and install a summer air-conditioning system, all to suit the requirements of three prospective long-term tenants: a furrier, a dentist, and a beautician. These changes are to be made on the first floor only, although unrestricted space is available in the basement for necessary equipment and air ducts.

Before beginning calculations for work of this type, it is important that the engineer set forth clearly what results the completed installation is to accomplish; under what conditions it is to operate, including any special or unusual requirements; and generally by what means the results are to be obtained. These are often called the "design assumptions." An understanding of these assumptions at the outset by both the designer and a responsible representative of those who are to benefit by his efforts may contribute greatly to the success of the installation. Many of the assumptions must be based upon purely arbitrary agreements on such matters as the indoor temperatures that are to be maintained under various conditions, the maximum number of lights that may actually be in use at any one time, or whether awnings will be lowered to ward off the direct rays of the sun. Other empirical assumptions must be made on the engineer's part, where exact data are lacking.

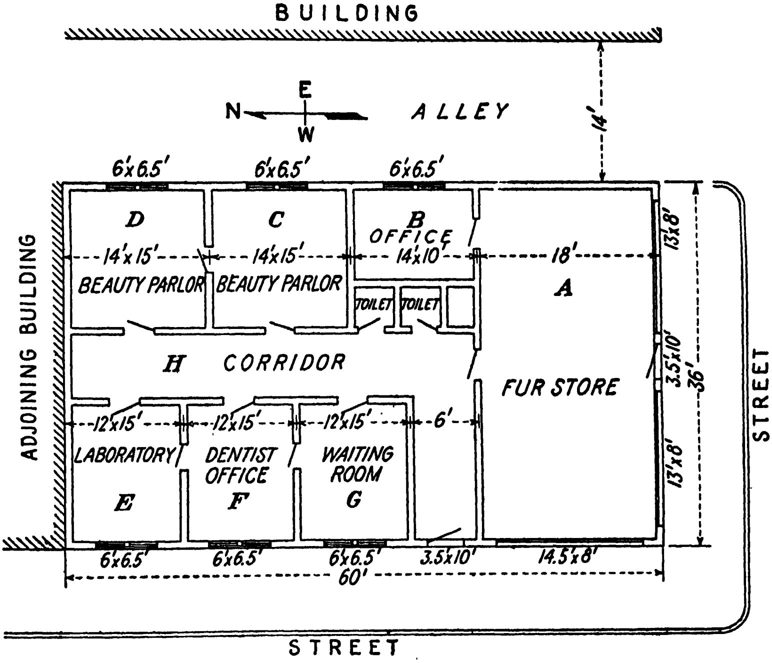

Fig. 304 —Plan of space to be air-conditioned.

368. Heating.—Since each room in this space is to be heated by a radiator, the heating calculations may be treated separately from the air conditioning. The following design assumptions may be made:

1. Each room to be maintained at 70° when the outdoor temperature reaches −10°. In this case a design temperature 14° above the lowest recorded is used (−24° + 14° = −10°).

2. Steam will be continuously available at the radiators at 2 lb. per sq. in. saturated.
3. The upper floors and the adjoining building to the north will be heated to 70° in winter.
4. Radiators in dentist's office and beauty parlor to require as little space as possible because of the small size of the rooms and to be convector type. In the fur shop and corridor cast-iron radiators are to be used.
5. Temperature control to be automatic.

Table 100 shows a complete summary of the calculations involved. To explain these, follow the steps taken in arriving at the required radiator surface for room *A* in Fig. 304. This room has three exposed sides, two of which are made up largely of glass. The exterior masonry walls consist of 4 in. of brick veneer backed by 8 in. of hollow tile, with $\frac{3}{8}$-in. decorative plasterboard furred on the interior. Referring to Table 7 in the Appendix, note that the coefficient of transmission *U* for this wall is 0.24. By Table 16 of the Appendix, the coefficient for single glass is 1.13.

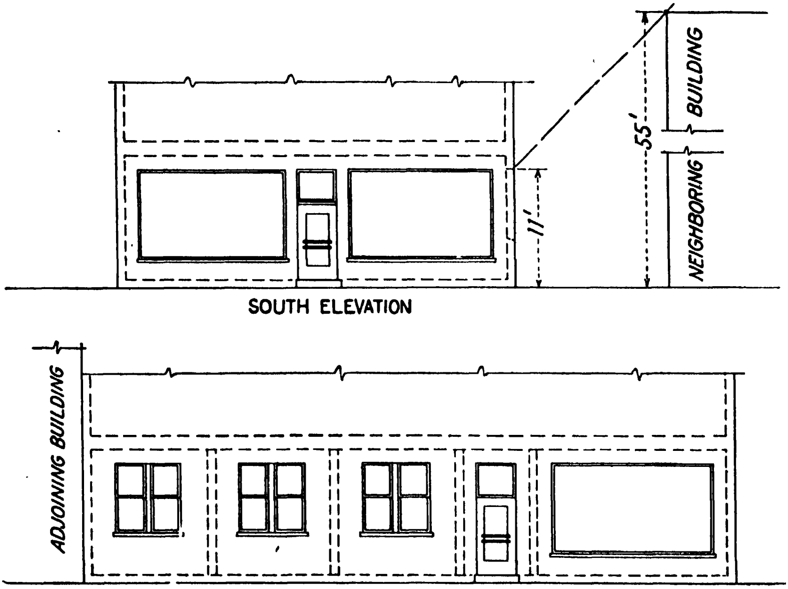

FIG. 305.—South and West elevations of space to be air-conditioned.

The area of exposed masonry wall may be most readily estimated by first finding the gross exposed area of the room and subtracting from it the area of the door and window openings. Thus:

Gross area = 12(18 + 36 + 18) = 864 sq. ft.
Glass area = (2 × 13 × 8) + (14.5 × 8) + (3.5 × 10) = −359
Net masonry wall area.................................. 505 sq. ft.

In arriving at glass area for this purpose, it is usual practice to treat the entire masonry opening for doors and windows as glass area, rather than endeavor to calculate coefficients for sash-frame materials.

Having established a maximum temperature difference between indoors and outdoors as 80°, it is now possible to calculate the corresponding heat loss through walls and windows. Thus:

Heat loss through wall = 505 sq. ft. × 0.24 × 80° = 9680 B.t.u. per hr.
Heat loss through glass = 359 sq. ft. × 1.13 × 80° = 29,600 B.t.u. per hr.

Since the windows in this room are sealed in fixed frames, the window-crack method of estimating infiltration is not applicable. Rather, an infiltration rate of two air changes per hour is chosen for a store of this type, which has a low rate of traffic through the entrance. The heat provided for this purpose must therefore equal that required to warm two room volumes of air from −10 to 70° in 1 hr. Thus:

Infiltration loss = 2(18 × 36 × 12) cu. ft. × 0.075 lb. per cu. ft. × 0.24 × 80
= 22,400 B.t.u. per hr.

The total heat loss from the room is

Wall	9,680 B.t.u. per hr.
Glass	29,600 B.t.u. per hr.
Infiltration	22,400 B.t.u. per hr.
Total	61,680 B.t.u. per hr.

The equivalent direct radiation required is therefore

$$\frac{61{,}680}{240} = 258 \text{ sq. ft.}$$

Wherever possible it is best to locate radiators beneath windows, and since in this case the window sills are only 24 in. above the floor, so-called "window-type" cast-iron radiators must be used. For approximate ratings of this type of radiator note in Table 1 of the Appendix that 20-in. sections of six-column radiator have a capacity of 5 sq. ft. E.D.R. Thus $^{258}/_{5} = 51.6$,

TABLE 100.—CALCULATION OF RADIATOR AND CONVECTOR SIZES

Room	Source of heat loss	Area or ft. of crack or rm. vol.	Coeff. U or cu. ft. per hr. per ft. crack	Temp. diff., °F.	Heat loss, B.t.u. per hr.	E.D.R.* req'd, sq. ft.	Equipment selected
A	Wall........	505	0.24	80	9,680		C.i. radiators (2) 20", 6 col., 18 sec (1) 20", 6 col., 16 sec.
	Glass.......	359	1.13		29,600		
	Infiltration..	7770†	2changes /hr.		22,400		
	Total.....				61,680	258	
B	Wall........	137	0.24	80	2,630		Convectors (2) 18" high × 28" long, top opening, sloping top
	Glass.......	39	1.13		3,530		
	Infiltration..	44‡	23.6§		150		
	Total.....				6,310	27	
C	Total.....	Same as *B*	Same as *B*		6,310	27	Same
D	Total.....	Same as *B*	Same as *B*		6,310	27	Same
E	Wall........	81	0.24	80	2,020		Same
	Glass.......	39	1.13		3,530		
	Infiltration..	44‡	63.4‖		403		
	Total.....				5,953	24	
F	Total.....	Same as *E*	Same as *E*	Same as *E*	5,953	24	Same
G	Total	Same as *E*	Same as *E*	Same as *E*	5,953	24	Same
H	Wall........	37	0.24	80	710		C.i. radiators (2) 38", 3 col., 24 sec.
	Glass.......	35	1.13		3,160		
	Infiltration..	600 c.f.m.			52,200		
	Total.....				56,070	234	

Total heat loss, first floor..................154,539 or 52 sections of this radiator are required. These should be grouped into three radiators as nearly as possible in proportion to the size of the windows under which they are located, say two 18-section radiators on the south and one of 16 sections on the west.

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* Equivalent direct radiation.

† Room volume, cu. ft.

‡ Cu. ft. per crack.

§ Cu. ft. per hr. per ft. crack—15 m.p.h. wind.

‖ Cu. ft. per hr. per ft. crack—30 m.p.h. wind.

Selection of radiation in the other rooms, as shown in Table 100, was made on a similar basis, except that convectors were used. Manufacturers of convectors rate them in a manner similar to cast-iron radiators, with added ratings of heat output for various heights of casing above the heating element.

369. Automatic Radiator Control.—There was a choice among the three types of heat control that might have been used on these radiators: (*a*) The pneumatic type, in which a thermostat varies compressed-air pressure on a diaphragm-operated radiator steam valve; (*b*) the electric type, in which a thermostat positions the rotor of a motor-operated valve; (*c*) the self-contained vapor-actuated type, in which a volatile liquid in a bulb-and-tube system expands or condenses in response to temperature changes, actuating a diaphragm-operated valve. These are all of a modulating type, that is, capable of holding a steam valve in intermediate or throttling positions and, generally, are all satisfactory. Since this was a rebuilding job, the vapor-actuated valve control was chosen. It is largely self-contained, with the bulb mounted on the wall close to the valve and under the convector intake. The pneumatic and electric types would have required runs of compressed-air piping or wiring between the valve and the thermostat location. These would have been unsightly if left exposed and rather expensive if concealed.

370. Cooling.—Several schemes for cooling this space in summer might have been used, such as (*a*) an individual cooling cabinet for each room, (*b*) several large cooling units supplying cold air through ducts to groups of rooms, and (*c*) a central cooling unit supplying the entire floor. Since in this case rental space was at a premium and the area was conveniently near to space in the basement where the apparatus could be located, the last of these schemes was chosen.

It was agreed that the design should be made on the basis of the following assumptions:

1. The capacity of the system shall be based on temperature representing an average of hourly records of outdoor conditions

for a large number of hot days in this locality, together with a corresponding indoor temperature as follows:

Time	Outdoor		Indoor dry bulb
	Dry bulb	Wet bulb	
9:00 A.M.	80	71	75
11:00 A.M.	85	72	75
3:00 P.M.	95	75	75
5:00 P.M.	95	75	75

The foregoing indoor temperature shall be maintained constant automatically *in each room* while the relative humidity of the indoor air is held at approximately 50 per cent.

2. The rented space will be in use until 6 P.M.

3. The second floor and basement and the adjoining building to the north shall be assumed to be window-ventilated and at outdoor temperature.

4. All windows when exposed to the sun will have awnings or Venetian blinds lowered. The Venetian blinds will have slats tilted 45 deg.

5. The installed lighting will be 100 per cent in use in any one room at the time of maximum heat gain and 65 per cent in use on the entire floor. Lighting wattage will be as shown in Table 102.

6. The number of occupants in each room will be as shown in Table 102.

7. The beauty parlor will have shampoo facilities and two 600-watt permanent wave machines in room *D*. In room *C* will be four 600-watt hair driers. All this equipment may be in operation simultaneously.

8. A toilet exhaust fan withdraws air from the two washrooms at the rate of 1200 c.f.m. This air comes from the corridor and enters the washrooms through louvers in the doors.

9. The system is to be designed so that rooms can be cooled with outdoor air without the aid of mechanical refrigeration, when the outdoor temperature drops to 60° or below.

371. Analysis of Cooling Load.—In order to proceed with the design, it is first necessary to separate the load into its component parts. These come under two main classifications, namely, the interior load and the exterior load.

The interior load consists of all the heat absorbed by the atmosphere in the space that is being cooled. The exterior load consists of all heat sources which, while they do not reach the space being cooled, do nevertheless appear as load on the cooling plant. Thus items of interior load are those due to (*a*) heat through ceiling, floor, and walls, (*b*) heat through windows, (*c*) heat of lights, (*d*) occupants, and (*e*) any heat-emitting apparatus that may be in the conditioned room. Items of exterior load are (*a*) heat removed in bringing fresh air down to the heat content of the air within the conditioned space, and (*b*) heat that recirculated air may pick up in returning from the conditioned space to the cooling apparatus. In addition to classifying these sources of cooling load, the designer must also ascertain what proportion of the heat to be removed from the air is represented by moisture or latent heat and what part by temperature change or sensible heat.

372. Computations for Cooling Load.—At various times of the day three sides of this building are exposed to the sun. To allow properly for variations in heat gain, due to the changing angle of the sun's rays and the rise of outdoor temperature during the day, it is first necessary to establish the heat-gain values of each exterior wall element for each exposure and at several significant hours of the day. In Table 101 all the data involved in arriving at these values for each square foot of exterior material are presented.

Direct exposure to the sun has an important effect on the cooling load in any room with windows, and it is equally important to recognize the effect of shade from permanent structures near by. In this case a substantial building stands immediately to the east and will cast a shadow along the entire east wall of the building in question during part of the morning hours. To allow for this accurately it is necessary to refer to tables[1] or curves[2] of solar altitudes and azimuths for the cooling months to be considered. In this case, however, it will be satisfactory to use August 1 as the design date. At that time solar altitude

[1] "American Ephemeris and Nautical Almanac."

[2] HOUGHTEN and others, "Heat Transmission as Influenced by Heat Capacity and Solar Radiation," *Trans. A.S.H. & V.E.*, 1932, p. 276, Fig. 36.

TABLE 101.—CALCULATION OF HEAT GAIN PER SQUARE FOOT OF WALLS AND GLASS

	Time of load effect	Time of load source	I	Temp. diff., °F.	U	F $0.23U$	a	B.t.u. per hr. per sq. ft.		
								H_t	H_R*	H
East wall, 4-hr. lag	9:00 A M.	5:00 A M	2.5	5	0.24	0.055	0.7	1.2		1.2
	11:00	7·00	2.5	5				1.2		1.2
	3:00 P.M.	11:00	94	10				2.4	3.6	6.0
	5:00	1:00 P.M	25.5	15				3.6	1.0	4.6
Glass, Venetian blinds, 58% I	9.00 A.M.		21	5	1.13			5.7	11.5	11.5
	11:00		94	10				11.3	54.5	54.5
	3:00 P.M.		21	20				22.6	12.2	22.6
	5:00		11	20				22.6	6.4	22.6
South wall			Assume same as east wall							
Glass, bare	9:00 A.M.		74	5	1.13			5.7	74	74.0
	11:00		124	10				11.3	124	124.0
	3:00 P.M.		74	20				22.6	74	74.0
	5:00		11	20				22.6	11	22.6
Glass, awnings, 28% *I*	9:00 A.M.		74	5	1.13			5.7	20.7	20.7
	11:00		124	10				11.3	34.7	34.7
	3:00 P.M.		74	20				22.6	20.7	22.6
	5:00		11	20				22.6	3.1	22.6
West wall, 4-hr. lag	9:00 A.M.	5:00 A.M.	2.5	5	0.24	0.055	0.7	1.2		1.2
	11:00	7:00	2.5	5				1.2		1.2
	3:00 P.M.	11.00 P.M.	25.5	10				2.4	1.0	3.4
	5:00	1:00	94.0	15				3.6	3.6	7.2
Glass, bare	9:00 A.M.		21.0	5	1.13			5.7	21.0	21.0
	11:00		25.5	10				11.3	25.5	25.5
	3:00 P.M.		195.0	20				22.6	195.0	195.0
	5:00		162.0	20				22.6	162.0	162.0
Glass, Venetian blinds, 58% *I*	9:00 A.M.		21	5	1.13			5.7	12.2	12.2
	11:00		25.5	10				11.3	14.8	14.8
	3:00 P.M.		195	20				22.6	113.0	113.0
	5:00		162	20				22.6	94.0	94.0
Glass, awnings, 28% *I*	9:00 A.M.		21	5	1.13			5.7	5.9	5.9
	11:00		25.5	10				11.3	7.2	11.3
	3:00 P.M.		195	20				22.6	54.6	54.6
	5:00		162	20				22.6	45.3	45.3
North wall, 6-hr. lag	9:00 A.M.	3:00 A.M.		5	0.16					0.8
	11:00	5:00		5						0.8
	3:00 P.M.	9:00		5						0.8
	5:00	11:00		10						1.6
Floor and ceiling, 2-hr. lag	9:00 A.M.	7:00 A.M.		5	0.25					1.25
	11.00	9:00		5						1.25
	3:00 P.M.	1:00 P.M.		15						3.75
	5:00	3:00		20						5.0

* According to the procedure outlined in Art. 353 for calculating heat gain H for windows, it is customary to use whichever is greater, H_R or H_t.

angles projected *in a vertical plane perpendicular* to an east wall are approximately as follows:

Sun time	8 A.M.	9 A.M.	10 A.M.	11 A.M.	12 A.M.
Sun angle	34°	42.5°	65.5°	82°	90°

By calculation, it may be determined that a line from the top of the first-floor east windows to the parapet of the neighboring building is 73 deg. above the horizontal. Thus the east windows will be exposed to direct sun rays, all or in part, from about 10:30 A.M. until noon.

In Table 101 are listed in two columns the solar intensities and outdoor-indoor difference variations on each exposed wall for four different hours of the day. The solar intensities may be found in Table 92, Chap. XIX, and the temperature differences in the design assumptions.

For the masonry walls a lag of 4 hr. was assumed between the time the heat entered the outside surface and the time it reached the inside surface. Thus the heat entering an east room at 3 P.M. was that due to the sun's rays and temperature difference affecting the wall at 11 A.M. From Art. 368 the coefficient U for this wall is found to be 0.24, and from the temperature schedule in the design assumptions, the difference between indoor and outdoor temperatures is 10° at 11 A.M. Thus the heat due to conductance at 3 P.M. is

$$H_t = 0.24 \times 10 = 2.4 \text{ B.t.u. per hr. per sq. ft.}$$

From Table 92 the solar radiation intensity I at 11 A.M. on east walls is 94 B.t.u. per sq. ft. The solar radiation factor F is taken as 0.23 U, which for wall is $F = 0.23 \times 0.24 = 0.055$. The outside surface of the wall is brick, and for this the solar absorption coefficient is $a = 0.7$. Assuming a 4-hr. lag, the solar heat entering the room through the wall at 3 P.M. is that which strikes the outer surface at 11 A.M. See Art. 352, Chap. XIX.

$$H_R = FaI = 0.055 \times 0.7 \times 94 = 3.6 \text{ B.t.u. per hr. per sq. ft.}$$

The total heat coming through a square foot of this section of the wall at 3 P.M. is

$$H = H_t + H_R = 2.4 + 3.6 = 6.0 \text{ B.t.u. per hr. per sq. ft.}$$

The heat coming through the north wall, floor, and ceiling is

TABLE 102.—CALCULATION OF HEAT GAIN IN ROOMS

Room	Source of heat	Wall or glass area, sq. ft.	9 A.M.		11 A.M.		3 P.M.		5 P.M.		Latent heat, B.t.u. per hr.
			B.t.u. per hr. per sq. ft.	B.t.u. per hr.	B.t.u. per hr. per sq. ft.	B.t.u. per hr.	B.t.u. per hr. per sq. ft.	B.t.u. per hr.	B.t.u. per hr. per sq. ft.	B.t.u. per hr.	
A	East wall	216	1.2	259	1.2	259	6.0	1,300	4.6	984	
	South wall	203	1.2	244	1.2	244	6.0	1,218	4.6	934	
	West wall	114	1.2	137	1.2	137	3.4	388	7.2	821	
	South glass, bare	21	74.0	1,552	124.0	2,610	74.0	1,552	22.6	474	
	South glass, awning	208	20.7	4,300	34.7	7,220	22.6	4,700	22.6	4,700	
	West glass, awning	104	5.9	614	11.3	1,175	54.6	5,680	45.3	4,710	
	Floor and ceiling	1296	1.25	1,490	1.25	1,490	3.75	4,860	5.0	6,480	
	Lights, 1600w			5,416		5,416		5,416		5,416	
	Occupants (4)			900		900		900		900	1060
	Total			14,912		19,451		26,014		25,419	1060
B	Wall	140	1.2	168	1.2	168	6.0	840	4.6	644	
	Glass, Venetian blinds	34	11.5	391	54.5	1,855	22.6	768	22.6	768	
	Floor and ceiling	280	1.25	350	1.25	350	3.75	1,050	5.0	1,400	
	Lights, 600w			2,045		2,045		2,045		2,045	
	Occupants (1)			225		225		225		225	265
	Total			3,179		4,643		4,923		5,082	265
C	Wall and glass same as *B*			559		2,043		1,508		1,412	
	Floor and ceiling	210	1.25	526	1.25	526	3.75	1,576	5.0	2,100	
	Lights, 600w			2,045		2,045		2,045		2,045	
	Occupants (5)			1,125		1,125		1,125		1,125	1325
	Hair driers, 2400w			6,135		6,135		6,135		6,135	2045
	Total			10,390		11,874		12,389		12,817	3370
D	Outer wall, glass, floor, and ceiling same as *C*			822		2,306		2,296		2,462	
	North wall	180	0.8	144	0.8	144	0.8	144	1.6	288	

	Lights, 600w			2,045		2,045		2,045		2,045	
	Perm. wave, 1200w			4,090		4,090		4,090		4,090	
	Hot-water evap										2100
	Occupants (4)			900		900		900		900	1060
	Total			8,001		9,485		9,475		9,785	3160
E	North wall same as *D*			144		144		144		288	
	West wall	144	1.2	173	1.2	173	3.4	490	7.2	1,038	
	Glass, Venetian blinds	34	12.2	415	14.8	503	113.0	3,840	94.0	3,196	
	Floor and ceiling	360	1.25	450	1.25	450	3.75	1,350	5.0	1,800	
	Lights, 600w			2,045		2,045		2,045		2,045	
	Occupants (1)			225		225		225		225	265
	Total			3,452		3,540		8,094		8,592	265
F	Outer wall, glass, floor, and ceiling same as *E*			1,038		1,126		5,680		6,024	
	Lights, 600w			2,045		2,045		2,045		2,045	
	Occupants (2)			450		450		450		450	530
	Total			3,533		3,621		8,175		8,519	530
G	Outer wall, glass, floor, and ceiling same as *E*			1,038		1,126		5,680		6,024	
	Lights, 600w			2,045		2,045		2,045		2,045	
	Occupants (5)			1,125		1,125		1,125		1,125	1325
	Total			4,208		4,296		8,850		9,194	1325
	Total sensible heat, all rooms			47,675		56,910		77,925		79,408	
	Less 35% heat of lights			−6,190		−6,190		−6,190		−6,190	
	Net total internal load			41,485		50,720		71,735		73,218	9975
H	North wall	72	0.8	58	0.8	58	0.8	58	1.6	58	
	West wall	51	1.2	62	1.2	62	3.4	174	7.2	368	
	Glass, bare	21	21.0	441	25.5	535	195.0	4,090	162.0	3,400	
	Lights, 900w			3,070		3,070		3,070		3,070	
	Total			3,631		3,725		7,392		6,896	

figured for conductivity only and with allowance for time lag as explained above.

The conductivity coefficient U for all sides of the conditioned space, except the north wall, may be found in the Appendix. However, the coefficient of the north wall, which consists of 13 in. of brick with ¾ in. of lath and plaster furred out on both sides, must be calculated in the manner explained in Art. 25. Thus:

$$U = \frac{1}{2\left(\frac{1}{1.9} + \frac{0.75}{3.3} + \frac{1}{1.9} + \frac{1}{1.9}\right) + \frac{13}{5.0}}$$

$$= 0.161 \text{ B.t.u per hr. per sq. ft.}$$

Windows having Venetian blinds transmit 58 per cent of the solar heat without time lag, and thus the heat entering 1 sq. ft. of glass at 3 P.M. on the west wall is

$$H_R = K \times I = 0.58 \times 195 = 113 \text{ B.t.u. per hr. per sq. ft.}$$

The time-heat transmission values having been assembled in Table 101, it is now possible to proceed with a complete tabulation in Table 102 of variations of cooling load in each room. Thus, room *B* has 140 sq. ft. of east masonry wall, which was previously found to transmit heat to the interior at 3 P.M. at the rate of 6.0 B.t.u. per hr. per sq. ft., or a total of 140 × 6.0 = 840 B.t.u. per hr. East glass with Venetian blinds at 3 P.M. transmits 22.6 B.t.u. per hr. per sq. ft., or for 34 sq. ft. in room *B* this is 34 × 22.6 = 768 B.t.u. per hr. The floor and ceiling were each found to transmit 3.75 B.t.u. per hr. per sq. ft. at 3 P.M. so that for a total of 280 sq. ft. the heat gain from this source will be 280 × 3.75 = 1050 B.t.u. per hr. The installed lighting wattage, which may be entirely in use at a time of maximum heat gain, is 600 watts, or 0.6 kw. The heat equivalent of a kilowatt is 3143 B.t.u. per hr., and for this room the lighting equals

$$0.6 \times 3413 = 2045 \text{ B.t.u. per hr.}$$

Room *B* will have one occupant, and with reference to Art. 240 the heat emission for an office worker moderately active is 225 B.t.u. per hr. sensible heat and 265 B.t.u. per hr. latent heat. In Table 102, all latent-heat values are tabulated separately. Latent heat might, of course, be reckoned as grains of moisture if preferred.

To find the cooling load in room *B* at 3 P.M., it is necessary to obtain the sum of all the loads that were computed above for that hour. Thus:

Room *B*, 3 P.M.	Sensible heat	Latent heat
Masonry wall	840	
Glass	768	
Floor and ceiling	1050	
Lighting	2045	
Occupants	225	265
Total B.t.u. per hr	4928	265

The foregoing example is typical for all rooms in Table 102, for loads at four different hours of a bright and relatively hot day. The next step is to find the total sensible heat gain for all rooms at each hour and the total latent heat gain.

Since, from the design assumptions, it is stated that for the area as a whole only 65 per cent of the installed lighting will be in use at a time of peak cooling load, the heat equivalent of 35 per cent of the lighting, or 35 per cent × 5.18 × 3413 = 6190 B.t.u. that must be deducted from each of the hourly sensible heat totals for all rooms.

The totals of hourly interior load do not include the values shown for heat gain in the corridor, because this space will not be cooled to the temperatures required in the tenants' rooms. However, the scheme of cooling to be adopted will involve using the corridor as a means of returning air from the cooled rooms to the air-conditioning equipment in the basement. It will therefore be necessary in later calculations to take into account the heat picked up in this return path.

The hourly interior loads having been determined, it is next necessary to calculate the exterior cooling loads for the same hours. A major item is the cooling required for outdoor air taken into the system. Since 1,200 c.f.m. of return air is withdrawn from the corridor and exhausted through the toilet rooms, it will be necessary to make up this same amount of air in the outdoor intake of the air-conditioning equipment. It will be found that this quantity of outdoor air more than satisfies the minimum ventilating requirements set forth in Art. 247.

According to the psychrometric chart, the air leaving the conditioned rooms at 75° and 50 per cent relative humidity has a specific volume of 13.67 cu. ft. per lb. and a wet-bulb temperature of 63°. The weight of air to be taken from outdoors is therefore

$$W = \frac{1{,}200}{13.67} = 87.8 \text{ lb. per min., or } 5{,}268 \text{ lb. per hr.}$$

In Table 66, the enthalpy of air at 63° wet bulb is found to be 28.48 B.t.u. per lb. The hourly differences of enthalpy of outdoor air and room air are therefore as follows:

	9 A.M.	11 A.M.	3 P.M.	5 P.M.
Wet bulb, outdoor air, °F	71	72	75	75
Enthalpy, outdoor air	34.83	35.70	38.46	38.46
Enthalpy, indoor air	28.48	28.48	28.48	28.48
Enthalpy difference	6.35	7.22	9.98	9.98

If each of these enthalpy differences is multiplied by the weight of outdoor air taken in, the result will be the exterior cooling load required for outside air. Adding to this the corresponding heat gain of return air in the corridor will give the total exterior cooling load. This is shown in the following tabulation:

	9 A.M.	11 A.M.	3 P.M.	5 P.M.
Heat removed, outside air	33,452	38,035	52,575	52,575
Heat from corridor	3,631	3,725	7,392	6,896
Total exterior load, B.t.u. per hr.	37,083	41,760	59,967	59,471
Total interior sensible heat per hr.	41,485	50,720	71,735	73,218
Total interior latent heat per hr.	9,975	9,975	9,975	9,975
Total load, B.t.u. per hr.	88,543	102,455	141,677	142,664

Also entered in the tabulation are the total hourly sensible- and latent-heat loads calculated in Table 102. An inspection of the total of all these heat sources shows the maximum load to occur at 5 P.M., and all further calculations may be based on conditions existing at that time.

373. Air Quantities.—With data available, it is now possible to determine the air quantity to be supplied to each room, the total amount to be handled by the air-conditioning apparatus, and the amount to be recirculated. The air quantity to be supplied to a given room is governed by the maximum rate at which sensible heat is to be removed and the maximum rise permissible in the supply air as it passes through the room. The required room temperature and the maximum rates of sensible heat gain have already been established. It will be assumed in this problem that 60° is the lowest permissible entering-air temperature, and that at this point its relative humidity is about 90 per cent. In room *A* the maximum sensible heat gain occurs at 3 P.M. and is 26,014 B.t.u. per hr. The air quantity required will therefore be

$$Q = \frac{(\text{B.t.u. per hr. } SH) \times \text{spec. vol.}}{60C_p(D.B.\text{ temp. rise in room})} \tag{1}$$

$$= \frac{26{,}014 \times 13.67}{60 \times 0.24(75 - 60)} = 1642 \text{ c.f.m.}$$

With this same procedure followed for each room, the individual air quantities required are found to be

Room	Max. sensible heat, B.t.u. per hr.	C.f.m.
A	26,014	1,642
B	5,082	321
C	12,187	780
D	9,785	618
E	8,592	543
F	8,519	538
G	9,194	581
Total....................		5,023

It will be seen that the total of these required room quantities is 5,023 c.f.m., which establishes the total air to be handled by the system. Special attention is called to the fact that this is a total of maximum air quantities required by each room, *regardless of the time* that each occurs.

Since the amount to be taken in from outdoors has already been determined, the recirculated air quantity is

$$Q_r = 5{,}023 - 1{,}200 = 3{,}823 \text{ c.f.m.}$$

or

$$W_r = \frac{3{,}823}{13.67} = 280.0 \text{ lb. per min.}$$

374. Air Circuit.—Figure 306 shows the schematic arrangement of the equipment to be used in this system. This is commonly known as the *double-duct system*, a type that will provide separate temperature control in each room, as required in the design assumptions. Its operation involves two supply-trunk ducts, one for carrying cold air, the other for warm. Each room is provided with a mixing damper, which releases varying proportions of cold and warm air from the two supply trunks in response to the demands of a thermostat in the room. The mixing-damper outlet is connected to a supply duct extending to the room air diffuser.

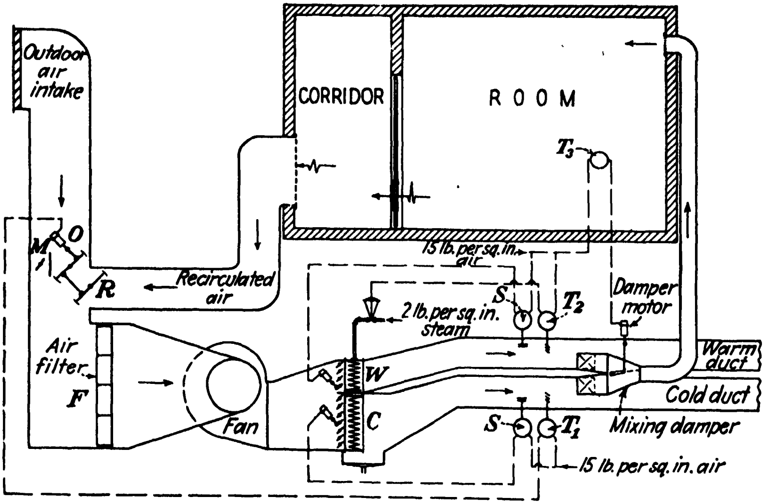

FIG. 306.—Scheme of air circulation and temperature control.

During hot weather *O* is closed and *R* is wide open, so that the fan draws air from both the return duct and the minimum-fresh-air damper *M* through filter *F*. Leaving the fan, the air passes through both the heating coil *W* (which is inactive) and the cooling coil *C*, to supply the demands of the mixing dampers from either trunk duct. After passing through the occupied rooms, the air leaves through return grilles provided in the door panels, to the corridor. A part of this air is exhausted through the toilet rooms, while the remainder flows into a return duct connected to the corridor.

During the cool weather of spring and autumn, when cooling may be required in sunny rooms and a small amount of heat may be needed in others at the same time, no refrigeration should be used. In that case coil *C* is inactive. Under these conditions dampers *O* and *R* proportion the quantities of outdoor and return air taken by the fan to maintain 60° at thermostat T_1. Steam is admitted to coil *W* in quantities necessary to maintain a temperature of 90° or less in the warm trunk, depending on the setting of thermostat T_2. Since this may involve recirculating only a small amount of air, provision must be made to relieve the excess returning to the corridor. It can be accomplished in this installation simply by opening the transom above the corridor entrance. In other instances it may be necessary to provide automatic relief dampers.

375. Thermal Calculations.—To give the manufacturer the proper specifications for performance requirements of the cooling coil, it is necessary to state the maximum air quantity it is to handle, the conditions of the air entering and leaving it, and the type of refrigerant that is to be used. Information on the rate of heat removal and the type and state of refrigerant required is also necessary in specifying the refrigerating equipment.

To obtain the condition of the air entering the coil, the sensible- and latent-heat content of both the return air and the outdoor air must be accurately determined, and for this purpose the values in Table 66 are used. The sensible heat of air leaving the rooms is that shown for dry air at 75°, or 17.99 B.t.u. per lb. Since this air is found to pick up 6896 B.t.u. per hr. in the corridor, the sensible heat gain is $\frac{6896\text{ B.t.u. per hr.}}{280\text{ lb. per min.} \times 60} = 0.41$ B.t.u. per lb. Thus the sensible heat of the return air entering the apparatus is $17.99 + 0.41 = 18.40$ B.t.u. per lb. The wet-bulb temperature of the return air is approximately 63°, and its enthalpy is the same as that for air saturated with water vapor at 63°, or 28.48 B.t.u. per lb. Thus the latent heat of this air is the difference between these two quantities, or $28.48 - 18.40 = 10.08$ B.t.u. per hr. In the same manner the sensible heat of 95° outdoor air is found to be 22.80 B.t.u. per lb., the enthalpy at 75°, 38.46 B.t.u. per lb.; and the difference, 15.66 B.t.u. per lb., is the latent heat.

The outdoor- and return-air quantities have already been established, and the proportion of each may be set forth as follows:

Outdoor air	87.8 lb. per min.	23.9 per cent
Return air	280.0 lb. per min.	76.1 per cent
Total	367.8 lb. per min.	100.0 per cent

The sensible and latent heat of the entering outdoor and return air mixtures may now be found as shown below:

	Sensible heat	Latent heat	Enthalpy of mixture
Outdoor air:			
0.239 × 22.80	5.45		
0.239 × 15.66	.....	3.74	
Return air:			
0.761 × 18.40	14.00		
0.761 × 10.08	.....	7.67	
Entering mixture	19.45	11.41	30.86

Referring again to Table 66 and interpolating, the dry-bulb corresponding to 19.45 B.t.u. per lb. sensible heat will be found to be 81.1°. Likewise the wet bulb corresponding to a saturated-vapor-mixture enthalpy of 30.86 is 66.2°. These two temperatures establish the coil entering air condition. When plotted on the psychrometric chart the entering air dew point will be found to be 58.2°.

Although the required rate of heat removal from the rooms has been found, two other sources of heat must now be considered before the required air conditions leaving the cooling coil are finally determined. These are the heat equivalent of the fan power exerted in circulating the air through the system, and the heat gained by transmission through the duct walls in unconditioned spaces. For convenience in this explanation, let it be assumed that the ducts have been designed so that the total static pressure required by the fan is calculated to be 1.5 in. water gage, and that by referring to fan tables it is found that the horsepower required is 1.5. Also let it be assumed that duct insulation has been selected that will limit the air temperature rise to 1.5°. The sensible heat gain between the apparatus and the room outlets will then be

Heat equivalent of power = 1.5 hp. × 2544 =	3,816 B.t.u. per hr.
Heat gain in ducts = 60 × 368 lb. per min. × 0.24 × 1.5 =	7,970 B.t.u. per hr.
Total	11,786 B.t.u. per hr.

The total sensible heat gained in the ducts and rooms per pound of air supplied is therefore

$$SH = \frac{73{,}218 + 11{,}786}{367.8 \times 60} = 3.85 \text{ B.t.u. per lb.}$$

Also, the total sensible plus latent heat gained per pound of air is

$$TH = \frac{(73{,}218 + 11{,}786) + 9{,}975}{367.8 \times 60} = 4.30 \text{ B.t.u. per lb.}$$

These two quantities are the difference of sensible and total heat contents between the air as it leaves the coil and as it leaves the room, or

	Sensible heat	Enthalpy of mixture
Heat of air leaving room	17.99	28.48
Heat gain of supply air	−3.85	−4.30
Heat of air leaving coil	14.14	24.18

Referring to Table 66, it will be found that 59° dry bulb and 56.7° wet bulb are the conditions corresponding to 14.14 B.t.u. per lb. sensible heat and 24.18 B.t.u. per lb. enthalpy of mixture. These two temperatures establish the coil leaving condition. When plotted on the psychrometric chart, it will be found that the dew point of the leaving air is 55.1°.

Using Eqs. (7), (8), and (9) explained in Art. 366, Chap. XIX, for the number of coil-tube rows, the tube-surface temperature, and the refrigerant temperature, respectively, the total refrigerating load may be estimated if the following calculated data and assumptions are used:

Calculated data:

$$t_1 = 81.1^\circ \qquad t_2 = 59.0^\circ$$
$$dp_1 = 58.2^\circ \qquad dp_2 = 55.1^\circ$$
$$Q_t = 5023 \text{ c.f.m.} \qquad W_t = 367.8 \text{ lb. per min.}$$
$$d = \frac{1}{13.67} = 0.0732 \text{ lb. per cu. ft.}$$

Assumed data:

$$\text{Coil-face velocity} = 500 \text{ ft. per min.}$$
$$\text{Coil-face area} = \frac{5023}{500} = 10.02 \text{ sq. ft.}$$
$$W = \frac{367.8 \times 60}{10.02} = 2{,}200 \text{ lb. per hr.}$$
$$h_o = 1.1(0.0758 \times 500)^{0.6} = 8.85$$
$$A_s = 20$$
$$h_r = 325 \text{ (for Freon)}$$
$$R = 15$$

Substituting in Eq. (7),

$$\frac{8.85 \times 20 \times N}{0.24 \times 2200} = \log_e \left(\frac{81.1 - 58.2}{59.0 - 55.1}\right)$$
$$0.335N = \log_e 5.87$$
$$N = 5.28 \text{ tube rows deep}$$

Substituting again in Eq. (7), using $N = 6$,

$$\frac{8.85 \times 20 \times 6}{0.24 \times 2200} = \log_e \left(\frac{81.1 - 58.2}{t_2 - dp_2}\right)$$
$$\text{antilog}_e\ 2.012 = 7.48 = \frac{22.9}{t_2 - dp_2}$$
$$t_2 - dp_2 = 3.06^\circ$$

Plotting these values on a psychrometric chart as explained in Art. 366 and shown in Fig. 307, the actual leaving coil conditions are found to be

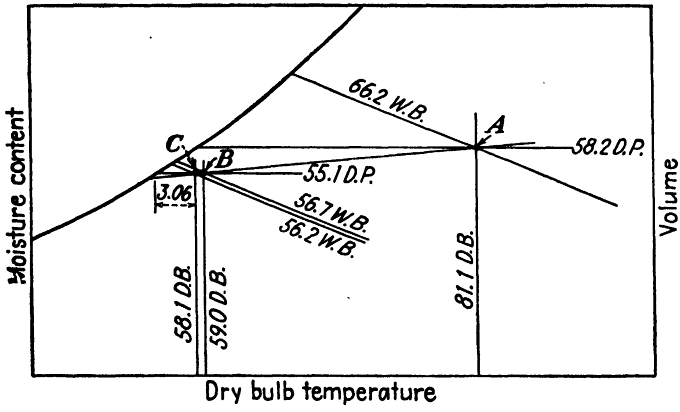

Fig. 307.—Graphical determination of leaving air conditions at coil.

58.1° dry bulb, 56.2° wet bulb, and 55.0° dew point. The enthalpy corresponding to this new wet bulb is shown in the tables as 23.90 B.t.u. per lb.

Since the enthalpies of the air entering and leaving the coil are now known, the required refrigeration may be determined as follows:

$$\begin{aligned} H &= W_t(h_1 - h_2) = 367.8(30.86 - 23.90) \\ &= 2560 \text{ B.t.u. per min.} \end{aligned}$$

or

$$T = \frac{2560}{200} = 12.80 \text{ tons of refrigeration.}$$

Substituting in Eq. (8), the coil surface temperature required for the above assumed conditions will be

$$\begin{aligned} t_s &= \frac{\left(\frac{81.1}{58.1} - \frac{58.2}{55.0}\right) 58.1 - 81.1}{\left(\frac{81.1}{58.1} - \frac{58.2}{55.0}\right) - 1} \\ &= 54.3° \end{aligned}$$

Likewise, the required refrigerant temperature is obtained with the aid of Eq. (9). In this case $H_t = \frac{2560 \times 60}{10.02} = 15{,}300$ B.t.u. per hr., and

$$\begin{aligned} t_r &= 54.3 - \frac{15{,}300 \times 15}{6 \times 20 \times 325} \\ &= 48.4° \end{aligned}$$

376. Coil Specification.—Since there is a wide diversity in manufacturers' designs of cooling coils, it is not customary to specify the number of tube rows, the amount of primary and secondary surface required, nor even the exact refrigerant temperature unless some special considerations require it. Rather, the air quantity, the entering coil condition, and the required leaving condition are specified, permitting the manufacturer to state in his proposal under what conditions his equipment will give the desired results. Thus for this example the coil specification may be written as follows:

> An extended-surface cooling coil which, when supplied with Freon-12 shall cool 367.8 lb. per min. of air entering at 81.1° dry bulb and 66.2° wet bulb to a leaving condition of 59° dry bulb and 56.7° wet bulb. The manufacturer shall supply the following information in his bid: (a) The quantity and temperature of refrigerant required, (b) the air-friction drop in inches of water when the coil surface is wet, and (c) complete information as to dimensions and materials used.

Circumstances will govern the writing of such a specification. This might be termed a relatively small installation and falls naturally into the class most adaptable to the use of the direct-expansion cooling coil. Building codes are severely limiting the size of such installations, however, as the possibility of the leaking of large quantities of refrigerant in the duct system may be a serious hazard. To overcome this objection in large installations, the refrigerant is limited to a heat exchanger that chills water. The water in turn is pumped in a closed circuit between the heat exchanger and cooling coils designed for use with chilled water. A similar situation would affect the selection of cooling coils in a system in which well water was used.

In direct-expansion systems it is often advantageous, in order to avoid divided responsibility, to have a contractor furnish and install the complete refrigerating system, including both the cooling coils and the refrigerating machine. Another important consideration in many direct-expansion systems, including the present example, is the possibility, during periods of light load, of the refrigerant's pressure and its temperature in the coils becoming too low. This will either cause the refrigerating machine to shut down automatically because of low suction pressure or to accumulate so much frost on the coil surface that air flow will be seriously obstructed. Certain manufacturers of multicylindered compressors have provided for such contingencies by an arrangement that cuts out cylinders in sequence automatically when the refrigerant temperature reaches predetermined limits. In effect this reduces the displacement of the compressor to match a wider range of load.

In view of the above considerations, the specification given for the cooling coil might be incorporated with the following refrigerating system specification:

> Furnish and install a refrigerating system (including a cooling coil with performance as specified above), all interconnecting piping, necessary automatic controls, and an initial charge of Freon-12 refrigerant. The compressor shall be equipped with provisions for automatic capacity cutout adapted to a cooling-load reduction of 50 per cent of maximum specified rating. The equipment shall be adapted to 220-volt 3-phase 60-cycle power and a maximum of 75° city water for condensing. The installation shall comply with the local refrigeration code in all respects.

377. Heating Coil.—As explained in Art. 374, a heating coil is required with this system in order to permit cooling certain rooms with outdoor air, while little or no cooling is required in others.

In actual practice it has been found that during cool weather, air supplied to an otherwise comfortable room must be delivered at a temperature several degrees higher than room temperature in order to avoid uncomfortable drafts. Thus heating coil *W* (Fig. 306), which under certain circumstances might have to handle the entire fan output, must be capable of warming this air from the fan outlet temperature to an arbitrary maximum of 85°. During this method of operation, dampers *O* and *R* may be maintaining 60° at the fan outlet for supply to the cold duct. This will also be the entering temperature to coil *W*.

The demand for warm air may vary widely from hour to hour with the positions of the mixing dampers in individual rooms. Consequently the flow of steam into the heating coil will vary to the same degree, if thermostat T_2 is to hold the warm trunk duct at a steady temperature. When steam flow is at a low rate, the tubes of a simple heating coil may be heated over only a small fraction of their length. This condition causes warm and cold air strata in the leaving duct, with resultant trouble in room-temperature control. Several coil manufacturers have provided ingenious designs for this that entail the use of a perforated tube within each heating tube of the coil. These perforated tubes serve to distribute the steam uniformly over the width of the coil at all rates of steam flow, thereby eliminating stratified leaving-air temperatures. This type of heating coil is to be recommended for the service required in the present example. If forced hot water were available for this purpose, a single-tube coil would be equally good, provided freezing temperatures were not to be encountered.

A specification for the heating coil required in this case may be written as follows:

> An extended-surface heating coil with internal steam-distributing tubes, having sufficient capacity to heat 5,023 c.f.m. of air entering at 60° to a leaving temperature of 85°. The face velocity[1] shall be approximately 600 ft. per min. Steam to the coil shall be throttled thermostatically between a maximum of 5 lb. per sq. in. gage saturated, to 0 lb. per sq. in. Air shall leave coil face at a uniform temperature at all rates of steam flow. Bidder shall state the air-friction drop through the coil in inches of water and provide complete information as to dimensions and materials used.

[1] "Face velocity" is that velocity corresponding to the flow through an area equal to the face area of the coil; its selection is arbitrary and based on the allowable air-friction drop through the coil.

378. Air Distribution.—It will be noted that the air quantities to each room have been based on the amount of 60° air required to absorb the maximum heat gain. During a great part of the time it follows that less air than this, or air of a higher temperature, will be required to prevent overcooling. Control of room temperature by varying air quantity has certain limitations, as explained in Art. 317, and may lead to inadequate ventilation and complaints of drafts when velocity from room outlets is varied. In this respect the double-duct system has particular merit, since the air quantity delivered to each room remains constant and its temperature is varied by means of a mixing damper. Obviously, in designing the warm and cold trunk ducts for this system, it must be assumed that either trunk may be called upon to deliver the entire design quantity.

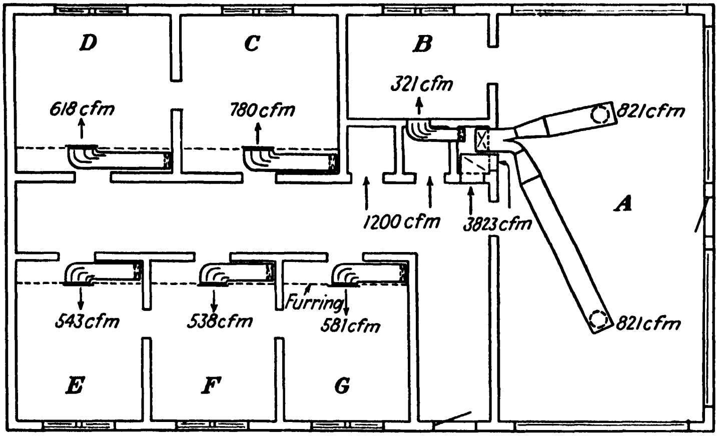

Fig. 308.—Air distribution on first floor.

Now that the air quantities to be handled in each part of the system have been determined, it is possible to proceed with the design of the ducts and the layout of the air-conditioning equipment shown in Figs. 308 and 309. These diagrams do not show the ductwork completely, since some duct length is required for the risers from the basement to the ceilings of the rooms. Table 103, however, gives the duct lengths involved in supplying one room and illustrates the method used in proportioning the ducts and in determining the total friction drop. All ducts were sized on a velocity basis with the particular objective of eliminating duct noise. In the trunk ducts the velocities become quite low at certain points, because the extra labor cost of transformation pieces or reducing fittings after each mixing-damper take-off more than offsets the slight extra cost in material of letting the duct run a short distance at uniform cross section.

Obviously it is not necessary to compute the total friction drop to each outlet in order to determine the total fan pressure required. Room *E* was selected because it is the longest run and thus requires the greatest pressure.

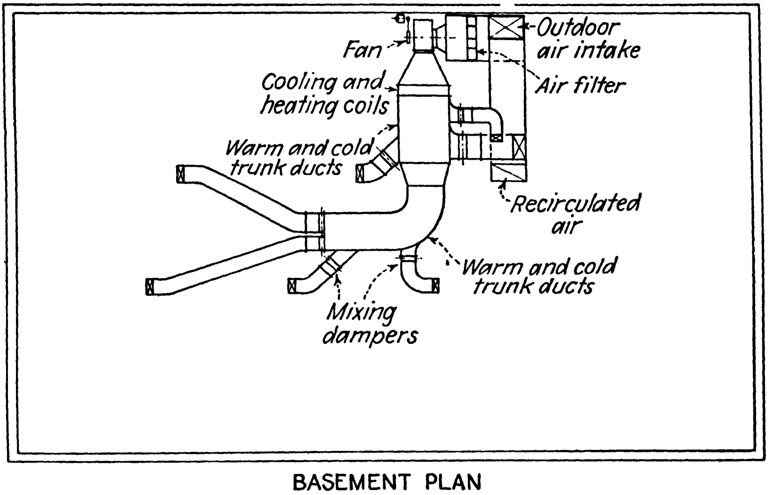
Fan
Cooling and heating coils
Warm and cold trunk ducts
Outdoor air intake
Air filter
Recirculated air
Warm and cold trunk ducts
Mixing dampers
BASEMENT PLAN

FIG. 309.—Arrangement of equipment and ducts in basement.

The diffusers selected for the small rooms in this installation were side-wall grilles fitted with vanes giving both horizontal and vertical direction control. The shape and face area used were based on the manufacturer's data applicable to the dimensions of each room under consideration. In making such selections, however, it is wise to provide more rather than less face area than is recommended, since it is a simple matter in case of draft complaints to block off effective face area from the rear of the grille. Pan-type diffusers were used in room *A*, since this type is adaptable to large areas, and it is assumed that the owner intends to provide a new dropped ceiling to conceal the ductwork.

379. Layout of Equipment.—Figure 309 shows the arrangement of all the equipment, including the necessary casings. A general

TABLE 103.—METHOD OF SIZING SUPPLY DUCT TO ROOM *E* AND DETERMINATION OF STATIC PRESSURE REQUIRED BY SUPPLY FAN

Obstruction or duct run	C.f.m. through duct	Duct size, in.	Duct area, sq. ft.	Equiv. round-duct diam.	Velocity, ft. per min.	Loss per 100 ft.	Duct length, ft.	Vel. head, in.	No. of vel. heads loss	Friction, in.
Pressure at diffuser...........	.....	......	(Design requirement)		...	.....	..	.....	....	0.100
Elbow, horizontal............	543	6 × 16	0.66	....	823	.....	..	0.042	0.70	0.029
Elbows, 2 vertical............	543	6 × 16	0.66	....	823	.....	..	0.042	0.13	0.011
Branch duct.................	543	6 × 16	0.66	10.5	823	0.11	30	.....	....	0.033
Mixing damper..............	.....	......	(Design requirement)		...	.....	..	.....	....	0.150
Trunk duct..................	2,280	12 × 36	3.00	22.2	760	0.035	14	.....	....	0.005
Elbow......................	2,280	12 × 36	....	....	760	.....	..	0.036	1.05	0.038
Transformation..............	2,280	......	....	....	760	.....	..	0.036	1	0.036
Trunk duct..................	5,023	18 × 48	6	31.6	837	0.035	10	.....	....	0.004
Trunk entrance..............	5,023	18 × 48	....	.....	837	.....	..	0.044	1	0.044
Cooling coil.................	5,023	(Manufacturer's data)			...	.....	..	.....	....	0.500
Pressure dampers............	.....	......	....	....	500	.....	..	0.015	1	0.015
Intake housing..............	.....	......	....	....	500	.....	..	0.015	1	0.015
Air filter...................	.....	(Manufacturer's data)			...	.....	..	.....	.....	0.400
Return damper..............	3,823	......	....	.....	850	.....	..	0.045	1	0.045
Return duct.................	3,823	18 × 36	4.5	27.7	850	0.035	15	.....	.....	0.005
Return elbow...............	3,823	18 × 36	....	....	850	.....	..	0.045	0.13	0.045
Return grille................	3,823	......	.....	....	850	.....	..	0.045	0.50	0.023
		Total pressure required by supply fan..								1.498

principle to be followed in such a layout is to avoid as many turns as possible. In this case, for instance, the axis of the air-filter housing is in line with the axis of the supply-fan intake, and the line of flow from the fan discharge is at right angles to the face of the cooling and heating coils. For most purposes centrifugal fans are available for either clockwise or counterclockwise rotation and for both horizontal and upward or downward discharge. A proper choice of these arrangements may greatly assist in making a good layout. Where a fan is to discharge through a coil and there is a considerable difference in dimensions between the fan outlet and the coil face, it is highly important that the two be well separated. It is then possible to connect them with a duct having a gradual flare, and a more uniform flow of air through the coil will result.

In arranging the cooling coil it should be borne in mind that when it is dehumidifying, the tube surfaces will be covered with condensed moisture, which will flow downward and off the coil by gravity or will be blown off in droplets. Provision must be made, therefore, to collect this water inside the housing and to drain it away. It may be necessary to install eliminator plates similar to those used in an air washer to stop the entrained moisture. The method used in this example was to mount the coil over a watertight pan with a drain at the bottom.

The velocities approaching the air filter should be low for proper performance, and the clearance around the filter bank should be ample for handling filter sections, since this is a frequent operation.

Thought must also be given to access to working parts inside the ducts and housings, such as fan bearings, control apparatus, and air filters, where inspection doors of generous dimensions should be provided. It is also wise to provide guards over fan intakes, and adequate inside light either in lamp fixtures or provisions for plugging in extension cords, both of which are important for safe operation.

380. Control.—The temperature control used in this installation has already been partially described. While self-contained control valves were chosen for the radiators for reasons explained in Art. 369, let it be assumed that 15 lb. per sq. in. compressed air is already available in the building for heating equipment in the upper floors. For that reason and because extensions to room thermostats will require a minimum of cutting and patching, pneumatic controls might be chosen.

The required automatic controls of this type have been indicated in Fig. 306. A gradual-acting room thermostat T_3 is necessary in each room to hold the mixing damper in intermediate positions by varying the air pressure applied to the damper motor. Duct thermostats T_1 and T_2 serve to position the diaphragm-operated steam valve and proportioning dampers O and R, maintaining the warm and cold trunk ducts at constant temperature. They are used only when cooling the building with outdoor air. These thermostats must also be gradual acting but should have low sensitivity to avoid hunting, *i.e.*, full travel between open and shut on small changes of temperature. A practical sensitivity for the room thermostat, however, will give full travel of the mixing damper from warm side to cold in a 1.5° change in room temperature.

A suitable type of refrigeration control has already been described in Art. 376. Controls of this type may or may not use a pneumatic thermostat. Satisfactory results are often obtained simply by controlling refrigeration from the temperature or pressure of the refrigerant returned to the compressor. Controls that stop and start the refrigerating machine to maintain constant temperature are generally not satisfactory with central fan systems.

Successful operation of the mixing dampers requires that the static pressure at both the warm and cold inlets be held constant and equal. For this reason static pressure controls are indicated on both trunk ducts. These act to position louver dampers over the faces of the heating and cooling coils. Where ducts of this type are long, it is best to locate the sensitive tip of the controller midway on the length where constant pressure is to be maintained, to avoid the effect of variations in duct friction with varying air flow.

The most important feature of this chapter is the procedure followed for the thermal calculations leading up to the coil specifications. There are other methods in use, but all are based on the same fundamental principles covered in Chaps. XIV and XIX. The student should follow through the mathematics sufficiently to obtain a clear conception of the principles involved.

The solution to this problem serves to illustrate what the writer believes to be the best current practice in the design of a central air-conditioning system involving a number of unrelated rooms. Actually, the student may see solutions to similar problems in which many features called for here have been omitted, and can judge for himself whether or not the best obtainable performance has been compromised. Although the same combination of requirements is seldom encountered twice, those presented are illustrative of the sort that make nearly every air-conditioning job a special one, calling for the exercise of ingenuity and judgement rather than a standardized application of set rules.

CHAPTER XXI

RESIDENCE AIR CONDITIONING

381. General.—Residence air conditioning is based on the same principles as the air conditioning of commercial buildings, but the apparatus used and its method of application have many peculiarities because of the nature of the service and the type of structure. In residence work two distinct fields have evolved —winter air conditioning, which is closely tied in with heating and includes filtering and humidifying; and summer air conditioning, which of course involves cooling and dehumidifying. Winter air-conditioning systems are much the more common, and in many localities an "air-conditioned house" means winter air conditioning only, although this is not always made clear to the layman. Summer air conditioning involves rather expensive apparatus which in the Northern states is used for but a relatively small portion of the year, and the majority of homeowners are not yet willing to pay the cost. In Southern states it is more fully appreciated and more frequently installed.

When summer air conditioning is installed, the apparatus is usually designed for winter operation also, and the system is therefore a year-round system. The homeowner, however, prefers to use window ventilation in spring and fall in preference to running a fan with windows closed during those months.

The winter air-conditioning system is usually based upon the mechanical warm-air furnace system and in fact is often simply a heating system of that type with means for cleaning and humidifying the air. The enormous popularization of the term "air conditioning" and the effort to develop satisfactory apparatus have resulted in a great improvement of the mechanical warm-air system, and the modern system does a very satisfactory job of heating when properly designed and installed. Unfortunately there have been many poor layouts and wrong applications.

382. Types of Residence Air-conditioning Systems.—The most common type of system is the **mechanical warm-air system** previously referred to. It is primarily a heating system and a winter air-conditioning system. When summer air conditioning is desired also, it requires simply the addition in the duct system of a cooling coil supplied by a refrigeration compressor, cold water from a well or the city supply, or water cooled by ice. The dehumidifying systems described in Chap. XVIII using silica gel or chemical absorbents may be used instead of mechanical refrigeration.

The warm-air heating apparatus is usually a **direct-fired** furnace, which is the cheaper method, or it may be an **indirect** system, consisting of a steam coil in the air duct and supplied by a steam boiler. These will be illustrated later.

Air conditioning with radiator heat is accomplished by installing a duct system—often for certain rooms only—which supplies filtered and humidified air and, if the cooling coil is added, cooled and dehumidified air. Such a system is quite analogous to the split system of ventilation or air conditioning often used in commercial buildings. It is a practicable way of adding air conditioning to an existing residence that has radiator heat.

Unit air conditioners, which are discussed in the next chapter, are frequently used in residences when it is desired to cool only a limited number of rooms. They are placed directly in the rooms to be cooled and are usually operated only in summer.

The **attic fan** is an important piece of equipment. It simply draws outdoor air into the house through open windows, especially at night. Because of its simplicity and low cost it has become popular. It accomplishes a good job of night cooling, in bedrooms particularly.

Mechanical Warm-air Systems.—Considered as a heating system only, the mechanical warm-air system is a marked improvement over the gravity warm-air system. If properly designed it ensures an adequate circulation of air to every room and therefore supplies reasonably uniform heat throughout the house. It uses smaller ducts, and because it does not depend upon gravity circulation the ducts can be installed close to the basement ceiling, making it possible to have a finished room in the basement, or in any case making the space more usable. The furnace need not be centrally located as with the gravity warm-air system.

Therefore even without the air-conditioning features the mechanical warm-air system has much in its favor. It is often

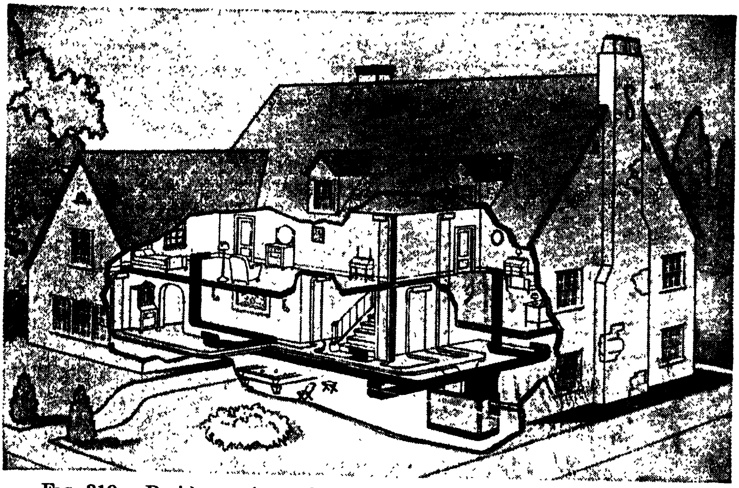

FIG. 310.—Residence air-conditioning system with direct-fired furnace.

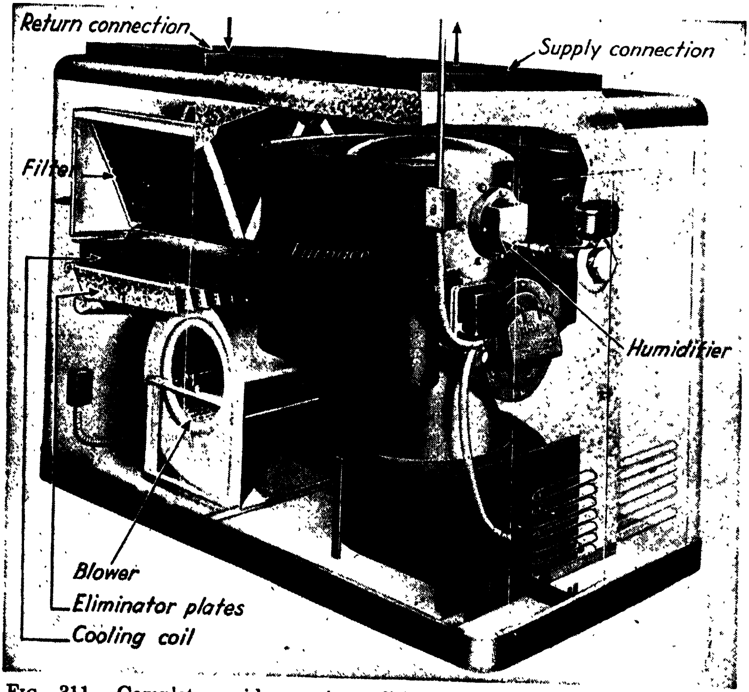

FIG. 311.—Complete residence air-conditioning apparatus. (*Timken Silent Automatic Division, The Timken-Detroit Axle Co.*)

chosen by the homeowner primarily because of those advantages rather than because of the air-conditioning features.

Figure 310 shows, somewhat pictorially, a system having a direct-fired furnace. The supply registers are high up in the walls, and the return registers are near the floor level.

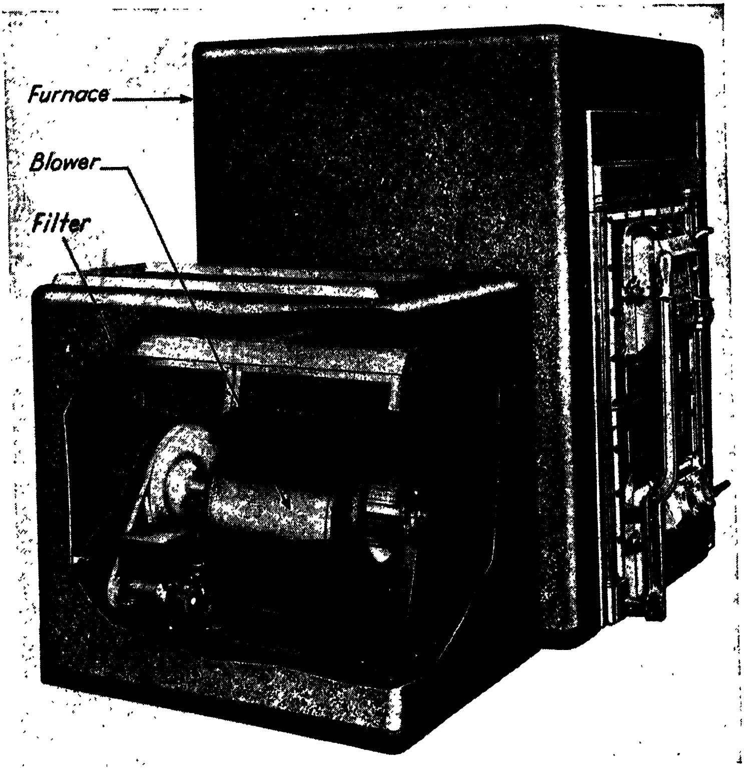

FIG. 312.—Hand-fired furnace with filter and blower. (*Fox Furnace Co., Elyria, Ohio.*)

Figure 311 shows a complete residence air-conditioning assembly, consisting of an oil-fired warm-air furnace with blower, filter, humidifier, and cooling coil. It is a year-round system, providing all phases of air conditioning. There are many variations in the design and arrangement, but the illustration is typical. The cooling coil is supplied with well water, city water (if below 60°), ice-cooled water, or refrigerant from a mechanical refrigerating unit.

Figure 312 shows a less elaborate furnace, coal-burning and hand-fired. It is equipped with a blower and filter.

An indirect system is shown in Fig. 313. The boiler supplies steam to the coil in the air-conditioning unit, which also contains the fan, filter, and humidifier. The chief advantage of this method is that radiators may be used in some rooms, such as the kitchen, bathrooms, and any rooms that may be isolated and consequently difficult to heat uniformly. Also, the domestic hot-water supply can be heated all the year by the boiler, whereas with the direct-fired furnace a separate water heater must be used, at least in the summer.

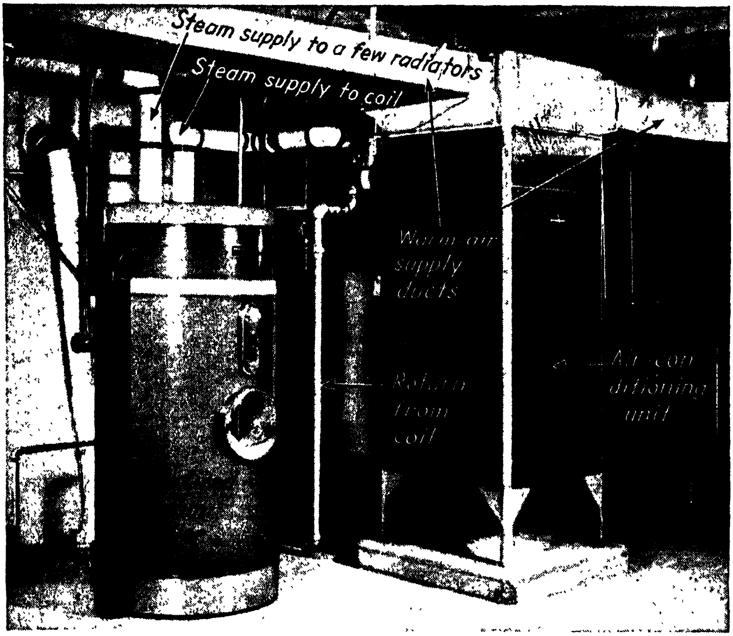

FIG. 313.—Indirect warm-air system with steam boiler. (*General Electric Company.*)

The indirect system is naturally more costly than the direct-fired system and is employed principally in larger houses. Hot water can be used instead of steam. The method of design of the ductwork is similar to the method used for furnace systems. It is of course important to select a coil that is adequate for the required air temperature.

383. Design Procedure.—The procedure developed by the National Warm Air Heating and Air Conditioning Association is generally recognized as the standard for designing a mechanical warm-air system. Until 1944 the Technical Code[1] served as a useful basis for the design and installation of such systems, and when properly interpreted and followed gave desirable results.

In 1945 the association adopted a new Code and Manual,[2] which simplifies the entire design procedure by utilizing various tabular data that list recommended combination units for the heat requirements of each room. For each *combination unit* a specific stack, branch pipe, and the register size are listed that will satisfy the required conditions. In addition the Code attempts to correlate better the selection and installation of registers, ductwork, furnace, blower, and controls, so that a completely coordinated system may be realized.

Although the latest Code eliminates such terms as "register temperatures," "air changes per hour," and "room-air quantity," none of these factors has been neglected. In fact, the latest research data available are incorporated in the design calculations that form the basis for the compilation of the tables.

For instance, over a period of years it has been generally recommended that air circulation equivalent to from five to six air changes per hour was required in a space to produce uniform temperature conditions. Recent tests in the association's research residence have indicated that six air circulations per hour did not give any more desirable results than three air changes. However, this same investigation did indicate that it was more desirable to have a smaller quantity of air circulated continuously than a larger quantity circulated intermittently.

The following paragraphs give in substance or by abstracted excerpts[3] the Code method of design.

Scope.—This Code is intended to apply to the design of furnace systems, having design heat losses up to 150,000 B.t.u. per hr., and for furnace-blower combinations having the following performance characteristics: (*a*) temperature rise through furnace not in excess of 100° and (*b*) static pressure available for overcoming pressure loss in distribution system of 0.20 in. water gage.

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[1] Technical Code for the Design and Installation of Mechanical Warm-air Heating Systems, 4th ed., Jan. 1, 1942, National Warm Air Heating and Air Conditioning Association, 145 Public Square, Cleveland, Ohio.

[2] Code and Manual for the Design and Installation of Warm-air Winter Air Conditioning Systems, 1945, N.W.A.H. & A.C.A.

[3] Printed in smaller type.

Heat-loss Calculations.—The first step in the design is the calculation of the heat losses for the various rooms. The heat-loss calculation in the Code is substantially like the method discussed in Chap. II, which is applicable for design purposes.[1] The furnace capacity is based upon the sum of all the room heat losses, using a formula that allows for a 15 per cent loss between the furnace bonnet and the registers.

384. Furnace Selection.—At the present time there are in use, or in a formative stage, several performance codes[2] for the testing and rating of mechanical warm-air furnaces. It is quite probable that all furnaces will eventually be tested under specified laboratory conditions. For furnaces not covered by existing codes, and particularly for coal-fired furnaces that have not been tested in accordance with Commercial Standard CS 109-44, the N.W.A.H. and A.C.A. has adopted (December, 1944) the following rating formulas applicable to forced-air furnaces.

For hand-fired coal furnaces converted to stoker, oil, and gas firing.

$$\text{Register delivery} = 2265 \times \text{H.S.} \tag{1}$$

For hand-fired coal furnaces with ratios of heating surface to grate area between 15 and 25:

$$\text{Register delivery} = 2265 \times \text{H.S.} \tag{2}$$

For hand-fired coal furnaces with ratios heating surface to grate area in excess of 25:

$$\text{Register delivery} = 2265 \times 25G \tag{3}$$

where Register delivery = output of furnace at register, B.t.u. per hr.

H.S. = total direct and effective indirect heating surface, sq. ft. (method of evaluating fin surface same as in Art. 64, Chap. IV)

G = grate area, sq. ft.

Bonnet capacity = register delivery × 1.18.

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[1] For any installation that is in dispute, the latest issue of the Code should be consulted and its method of heat-loss calculation followed meticulously.

[2] American Standard Approved Requirements for Central Heating Gas Appliances, A.S.A.; Warm Air Furnaces Equipped with Vaporizing Pot-type Burners, Nat. Bur. Standards Commercial Standard CS 104-43; Solid-fuel Burning Forced Air Furnaces (up to 80,000 B.t.u. per hr.), Nat. Bur. Standards Commercial Standard CS 109-44; Tentative Code for Testing, Oil-fired, Fan-furnace Units, 6th ed. (1941), N.W.A.H. & A.C.A.

In deriving the above equations, consideration was given to register air temperature, flue-gas temperature, metal surface temperature, draft, and combustion rate. Admittedly, any rating derived from Eq. (1), (2), or (3) is not so reliable as a rating determined from actual tests. However, these formulas do have a rational basis for their derivation and are more accurate than any equations commonly used by industry in previous years.

385. Register Location.—The next step in the design involves locating on a plan sketch of the structure an indication of the location of warm-air registers and return-air openings. Any consideration of the type and location of registers also requires consideration of the method of blower operation and controls utilized. Simple basic rules for the proper installation of registers are presented herewith.

High Side-wall Registers are extensively used and have many advantages, as follows: (1) there is practically no danger of the air stream striking an occupant, (2) they do not interfere with furniture placement, and (3) they are best adapted to summer cooling.

With practically continuous blower operation, floors are kept warm, since the cool air is drained continuously. Intermittent blower operation that results in long off periods of the blower will not give satisfactory operation. In general, warm-air registers should be located in warm walls so that the air discharges in the direction of cold exposed walls.

Adjustable two-way deflecting-type registers, as shown in Table 108, are **preferred types.** Registers are usually located with the bottom of the opening about 6½ ft. from the floor, and with the top of the opening not less than 1 ft. below the ceiling.

Where the register is closer than 1 ft. to the ceiling, a slight downward deflection, not in excess of 15 or 22 deg., is desirable. Where the register is in the center of the side wall, deflect the air stream sideways to both sides. Where the register is near a wall, deflect the stream away from the wall. Register sizes shown in Table 108 are **industry standards.**

Low Side-wall and Baseboard Registers require care in selecting the proper location. When properly located and properly selected, they are particularly effective in counteracting drifts of cool air due to exposed walls or to leakages from doors and windows, and in attacking cold at the source of leakage.

Air that is discharged against the base of a stairway will break up any descending currents of cool air. Air that is discharged near a front door will tend to neutralize the cold air that enters the house from leaky doors or when the door is opened.

Some installers prefer baseboard registers in basement rooms with bare concrete floors, in order to blanket the floor with warm air.

The living room is the key room in the house. Particular attention must be given in locating registers in this room so that the air stream will not discharge against an occupant at rest.

Adjustable-type registers with vertical and horizontal deflection are recommended. Use both sideways-deflection- and downward-deflection-type registers to dissipate the velocity of air rapidly. See Table 108 for preferred types of registers. The sizes shown in Table 108 are industry standards.

Registers are usually located on inside walls.

Ceiling Registers give results similar to those of high side-wall registers. They must be of a type that provides sideways deflection. Deflect the air toward colder wall surfaces and away from places where people will be sitting. Ceiling locations are better adapted to rooms having high ceilings.

Floor Registers are satisfactory for installation under large glass exposures, such as bay windows, where the ascending warm air will counteract the descending drift of cool air from window surfaces.

Floor registers are ordinarily not preferred, since they interfere with furniture placement and rugs and curtains. In old house construction, existing floor registers may often be retained, if construction limitations do not permit the use of new register locations.

Return-air Intake Locations.—Intakes can be located either on inside or outside walls. Where large sources of cold air exist, the intakes are usually located near such sources. Intakes should be located so as to permit a simple, direct ductwork. Air from a room should be removed from floor level. Hence, intakes should not be located above the baseboard.

When draperies extend down to the floor level, do not place the return intakes below the windows where the draperies will cover them when fully drawn. Place the intakes at either side of the window, or if permissible, in the floor below the window.

Return intakes are usually located in all rooms except bathrooms, closets, and lavatories.

In a two-story house where an open stairway goes up from the living room, it is advisable to place a return in each second-story room, of sufficient capacity to balance the warm-air supply. If the stairs go up from an entrance hall, a substantial portion of the return air from the second story can be handled by a return intake in the lower hall. If floors on a given story are on different levels, one or more returns must be provided at the lowest level.

Locate and size the intakes so that an excessive amount of return air is not drawn across the living-room floor.

Blower Operation.—A building is most satisfactorily heated by a winter air-conditioning system when the **blower operates for long periods** in mild weather, and practically continuously in weather colder than about 40°. Set the blower for the lowest possible speed to give the volume of air required to heat the house in the most severe weather. More uniform heating will be obtained by circulating a reasonable volume of air continuously than an excessive volume of air intermittently.

The heat input should be so controlled that in mild weather the **burner, or draft damper, operates frequently,** but only for short periods. In the case of automatic-fired equipment, adjust the fuel-burning equipment for proper fuel and air input to the furnace. By doing so, the temperature of the air that is delivered to the rooms is modulated to the actual heating requirements of the building for all ranges of weather.

The blower should operate **until the furnace is cooled;** otherwise heated air will circulate by gravity through the nearest registers or those highest above the furnace.

The **ideal method of operation,** therefore, is that in which the blower operates for prolonged periods and stops only when the temperature of the circulating air is so low that gravity circulation becomes negligible.

Controls.—When all the warm air registers are at the **high side-wall** location, the fan switch should be adjusted to start the fan when the bonnet air temperature has reached about 110°, and to shut off at about 25° lower than the starting temperature.

When any warm-air registers are located at the **low-wall, baseboard,** or floor locations, slightly higher air temperatures will usually be required. The fan switch should be adjusted to start the fan when the bonnet air temperature has reached about 130° or slightly less, and to shut off at about 25° below the starting temperature.

It is desirable that the blower be started when the bonnet air temperatures are low. The limit switch should be set, independently of the fan switch, so that the bonnet air temperature cannot exceed about 175°. Often the limit switch cannot be adjusted independently of the fan switch. In this case, set the fan switch at the lowest bonnet temperature that will heat the house in the most severe weather.

Where a blower is operating almost continuously and the heat source is operating intermittently, the **temperature of the air** that is delivered from the registers will vary from about 80 to 150°. Experience has proved that these temperature limits are satisfactory with proper selection and location of warm-air registers.

386. Selecting Warm-air Combinations.—Before proceeding to select actual sizes for the system layout, it is next desirable to locate on the floor and basement plans, such as in Figs. 314 and 315, the actual position of the warm-air registers and return-air intakes. After indicating all the warm-air and return-air risers on the basement plan, single-line diagrams should be constructed connecting all of the risers to separate supply and return trunk-duct systems. The following outline of procedure should then be followed to select the proper duct and register combinations:

1. Using the form in Table 104, enter on line 1 the name of the room in which the register is located. Indicate on line 2 the location of room with reference to elevation and on the next line record the heat loss previously calculated.

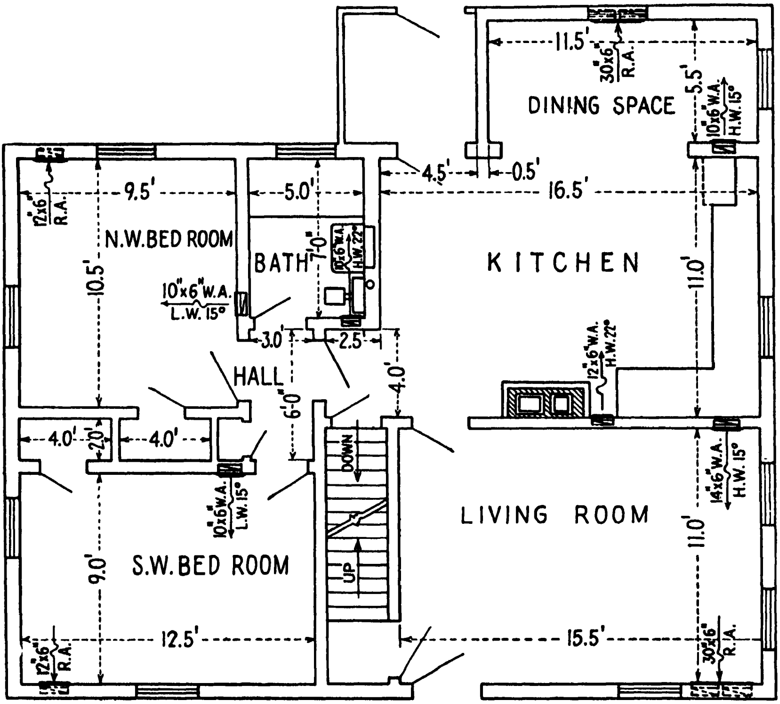

FIG. 314.—Typical example showing locations of registers and intakes.

2. Enter the actual length, in feet, on line 5, measured along the trunk and branch from the furnace bonnet to the stack or riser boot. Do not include lengths of vertical risers in actual length measurements, since these are accounted for in capacity tables.

3. Tabulate on line 6 the sum of the equivalent lengths of all fittings and the register appearing in each run. Obtain this information from Fig. 316 and Table 105. An example covering this procedure of tabulation is shown in Fig. 317.

TABLE 104.—DATA FOR SIZING WARM-AIR AND RETURN-AIR BRANCHES

1	Room use		N.W. bed		S.W. bed		Living		Kitchen		Dining		Bath & hall		Recreation	
2	Room No.	Story	1	1st	2	1st	3	1st	4	1st	5	1st	6	1st	7	Basement
3	Total room B.t.u. loss		5642		6922		13,433		8783		6113		6670		13,820	
4	The sum of all the room B.t.u. losses is the total building B.t.u. loss = 57,383 B.t.u. per hr.															
	Warm-air Branches															
5	Length bonnet to boot, ft.		19		18		13		8		15		13		31	
6	Equiv. length of fittings and register		120 eq. ft.		115 eq. ft.		130 eq. ft.		130 eq. ft.		130 eq. ft.		135 eq. ft.		110 eq. ft.	
7	Combination no.		42		42		44		43		42		41		Each 43	
8	Stack size, in.		10 × 3¼		10 × 3¼		14 × 3¼		12 × 3¼		10 × 3¼		10 × 3¼		12 × 3¼	
9	Branch (or leader) size, in.		6" rd.		6" rd.		6 × 8		7" rd.		4 × 8		4 × 8		5 × 8	
10	Register size, in.		10 × 6		10 × 6		14 × 6		12 × 6		10 × 6		10 × 6		12 × 6	
11	Register finish		Prime		Prime		Prime		Prime		Prime		Prime		Prime	
12	Register location		FL* BB† (LW)‡ HW§ CLG‖		FL BB (LW) HW CLG		FL BB LW (HW) CLG		FL BB LW (HW) CLG		FL BB LW (HW) CLG		FL BB LW (HW) CLG		FL BB LW HW (CLG)	
	Return-air Branches															
13	Room Nos. to be served		1 and 6		2		3		See 5		4 and 5		See 1		7	
14	Length plenum to boot, ft.		30		33		31		..		18		..		6	
15	Equiv. length of fittings only		100 eq. ft.		100 eq. ft.		150 eq. ft.		..		100 eq. ft.		..		80 eq. ft.	
16	Desired B.t.u. capacity		8312		6922		13,433		..		14,896		..		13.820	
17	Combination No.		53		53		56		..		55		..		53	
18	Stack size		Joist space		J.S.		2, J.S.		..		2, J.S.		..		12 × 3¼	
19	Branch duct size, in.		5 × 8		5 × 8		10 × 8		..		8 × 8		..		..	
20	No. of joists & depth req'd.		1 at 4 in.		1 at 4 in.		2 at 4 in.		..		2 at 3 in.		..		..	
21	Intake size, in.		12 × 6		12 × 6		30 × 6		..		30 × 6		..		12 × 6	

* Floor.
† Baseboard.
‡ Low side wall.
§ High side wall.
‖ Ceiling.

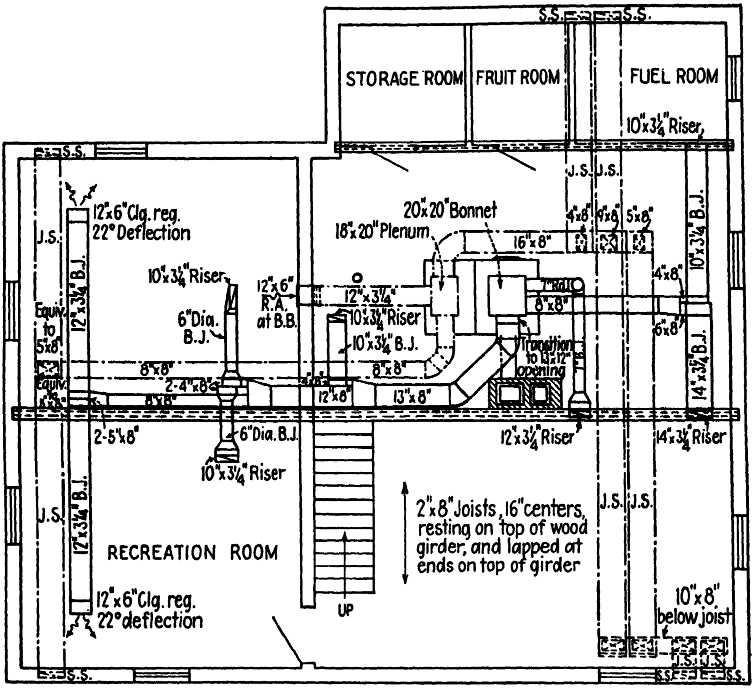

FIG. 315.—Typical duct plan.

TABLE 105.—EQUIVALENT LENGTH FOR REGISTERS
Including losses in stack head and velocity pressure

Deflection angle, deg.	Baseboard or low side-wall register, equiv. ft.	High side-wall register, equiv. ft. for throw of air	
		Less than 13 ft.	Over 13 ft.
Col. 1	Col. 2	Col. 3	Col. 4
0	35	35	70
15	40	40	80
22	45	45	90
30	60	60	115
45	155	115	230

NOTE.—When turning vanes are used in stackhead use 0.7 of values in columns 2 to 4. Values shown in columns 2 and 3 are for 6-in. registers (standard height); those in column 4 are for 4-in. registers. For 5-in. registers multiply values in columns 2 and 3 by 1.3. For 8-in. registers multiply values in columns 2 and 3 by 0.8.

For 2-way deflection registers add the vertical and horizontal deflection angles together and multiply by 0.7. Then select closest angle in column 1.

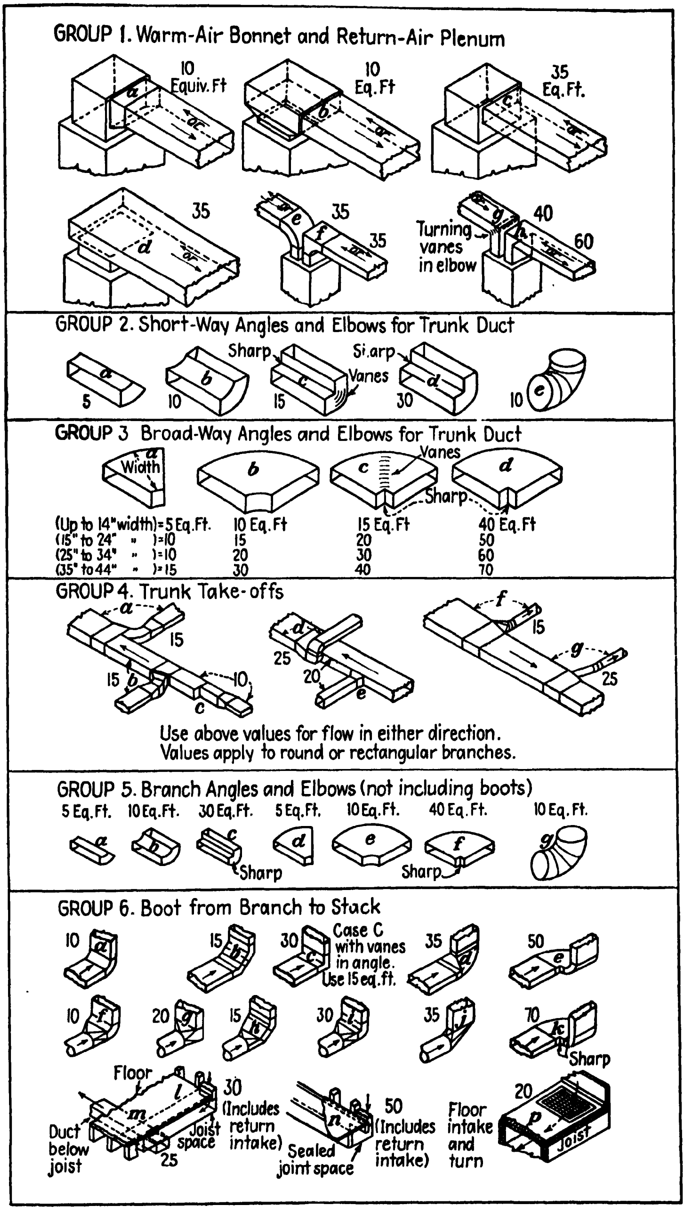

FIG. 316.—Equivalent length of fittings.

TABLE 106.—CAPACITY TABLES FOR FIRST-STORY WARM-AIR AND RETURN-AIR BRANCHES, B.T.U. PER HOUR

For uninsulated metal ducts	Actual length from bonnet to boot or from return plenum to boot, ft.								Warm-air combination number	Return-air combination number
	1–7	8–12	13–17	18–24	25–34	35–44	45–54	55–64		
	Col. *a*	Col. *b*	Col. *c*	Col. *d*	Col. *e*	Col. *f*	Col. *g*	Col. *h*		
Section *A*	7,200	6,700	6,100	5,600	4,800	4,100	3,500	3,000	41	51
40–69	12,500	11,700	10,800	9,900	8,500	7,400	6,400	5,500	42	52
Equiv. ft. for	16,000	15,000	14,000	13,000	11,300	9,900	8,700	7,700	43	53
fittings and	19,100	18,000	17,000	16,000	14,200	12,500	11,000	9,600	44	54
register	25,000	23,400	21,600	19,800	17,000	14,800	12,800	11,000	45*	55
	32,000	30,000	28,000	26,000	22,600	19,800	17,400	15,400	46*	56
	80,000	75,000	70,000	65,000	56,500	49,500	43,500	38,500	. . .	57*
Section *B*	5,500	5,100	4,800	4,500	3,900	3,400	3,000	2,600	41	51
70–99	9,900	9,200	8,600	8,100	7,100	6,200	5,400	4,600	42	52
Equiv. ft.	13,100	12,300	11,600	10,900	9,700	8,500	7,500	6,500	43	53
	16,300	15,400	14,500	13,700	12,200	10,800	9,500	8,300	44	54
	19,800	18,400	17,200	16,200	14,200	12,400	10,800	9,200	45*	55
	26,200	24,600	23,200	21,800	19,400	17,000	15,000	13,000	46*	56
	65,500	61,500	58,000	54,500	48,500	42,500	37,500	42,500	. . .	57*
Section *C*	4,400	4,200	3,900	3,700	3,200	2,900	2,600	2,300	41	51
100–139	8,100	7,600	7,100	6,600	5,800	5,100	4,500	4,000	42	52
Equiv. ft.	10,800	10,100	9,600	9,000	8,000	7,100	6,200	5,400	43	53
	13,700	12,900	12,100	11,300	10,000	8,900	7,900	7,000	44	54
	16,200	15,200	14,200	13,200	11,600	10,200	9,000	8,000	45*	55
	21,600	20,200	19,200	18,000	16,000	14,200	12,400	10,800	46*	56
	54,000	50,500	48,000	45,000	40,000	35,500	31,000	27,000	. . .	57*
Section *D*	3,900	3,700	3,500	3,300	2,900	2,500	2,200	2,000	41	51
140–189	6,900	6,500	6,100	5,700	5,100	4,500	4,000	3,600	42	52
Equiv. ft.	9,300	8,700	8,100	7,600	6,800	6,000	5,300	4,700	43	53
	11,600	11,000	10,400	9,800	8,700	7,700	6,800	6,000	44	54
	13,800	13,000	12,200	11,400	10,200	9,000	8,000	7,200	45*	55
	18,600	17,400	16,200	15,200	13,600	12,000	10,600	9,400	46*	56
	46,500	43,500	40,500	38,000	34,000	30,000	26,500	23,500	. . .	57*
Section *E*	3,300	3,100	2,900	2,700	2,500	2,200	1,900	1,700	41	51
190–250	5,700	5,400	5,100	4,800	4,300	3,800	3,400	3,000	42	52
Equiv. ft.	7,800	7,300	6,900	6,500	5,700	5,000	4,400	3,900	43	53
	9,700	9,100	8,600	8,100	7,300	6,500	5,800	5,100	44	54
	11,400	10,800	10,200	9,600	8,600	7,600	6,800	6,000	45*	55
	15,600	14,600	13,800	13,000	11,400	10,000	8,800	7,800	46*	56
	39,000	36,500	34,500	32,500	28,500	25,000	22,000	19,500	. . .	57*
For insulated ducts	Col. *a* 1–9	Col. *b* 10–17	Col. *c* 18–24	Col. *d* 25–34	Col. *e* 35–54	Col. *f* 55–74	Use these column headings for ducts that are completely insulated with ½-in. thick insulation from bonnet to boot			

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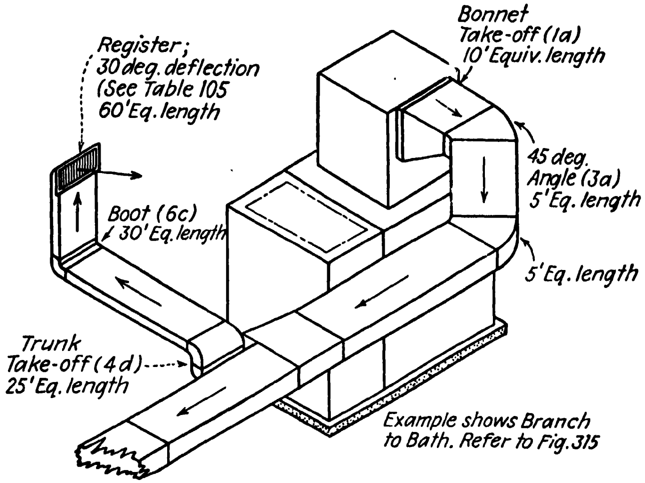
* Use these items only when the building construction or capacity requirements necessitate the use of two adjoining stacks or floor registers.

4. Select and enter on line 7 *combination numbers* from Table 106 for first-story and basement rooms. Use Table 107 for second-story rooms. This is done by locating the vertical column in Tables 106 or 107 containing the actual length of run shown in line 5. Locate the horizontal section containing the equivalent length, in feet, of fittings and register from line 6. Then note in the respective columns the B.t.u. capacity nearest to that required from line 3. In the same horizontal line with B.t.u. capacity, find the proper warm-air combination number and enter this on line 7.


Register; 30 deg. deflection (See Table 105 60' Eq. length
Bonnet Take-off (1a) 10' Equiv. length
45 deg. Angle (3a) 5' Eq. length
5' Eq. length
Boot (6c) 30' Eq. length
Trunk Take-off (4d) 25' Eq. length
Example shows Branch to Bath. Refer to Fig. 315

Bonnet take-off fitting, Type 1*a*	10 equiv. ft.
Broad-way trunk angle, Type 3*a*	5 equiv. ft.
Broad-way trunk angle, Type 3*a*	5 equiv. ft.
Trunk take-off fitting, Type 4*d*	25 equiv. ft.
Boot from branch to stack, Type 6*c*	30 equiv. ft.
Register, 30 deg. deflection, Table 105	60 equiv. ft.
Total equivalent length of fittings	135 equiv. ft.

FIG. 317.—Method of adding equivalent lengths of fittings.

If no B.t.u. capacity or combination number is found in the proper section, it is an indication that a single riser is not sufficient. In such cases the heat loss must be divided, and two or more branches or risers must be used.

5. From Table 108, tabulate for the proper combination number the stack size on line 8, the branch pipe or duct on line 9, and the register size on line 10. Line 12 is provided to check the register location with abbreviations given for floor, baseboard, low side wall, high side wall, and ceiling locations.

6. Capacity Tables 106 and 107 are based on the use of uninsulated galvanized-iron ducts and risers surrounded by 70° air. If part of the duct system passes through an unheated room, such portions should be covered with at least ½-in. insulation. It is good practice to insulate ducts that are more than about 25 ft. long. At the bottom of the capacity tables

TABLE 107.—CAPACITY TABLES FOR SECOND-STORY WARM-AIR AND RETURN-AIR BRANCHES, B.T.U. PER HOUR

For uninsulated metal ducts	Actual length from bonnet to boot or from return plenum to boot, ft.								Warm-air combination number	Return-air combination number
	1–7	8–12	13–17	18–24	25–34	35–44	45–54	55–64		
	Col. *a*	Col. *b*	Col. *c*	Col. *d*	Col. *e*	Col. *f*	Col. *g*	Col. *h*		
Section *A* 40–69 Equiv. ft. for fittings and register	6,300	5,700	5,200	4,800	4,100	3,500	3,100	2,800	41	51
	10,900	10,000	9,200	8,500	7,300	6,400	5,600	5,000	42	52
	14,000	13,000	12,100	11,400	10,000	8,800	7,800	6,800	43	53
	17,000	15,900	14,900	13,900	12,400	11,100	9,900	9,000	44	54
	21,800	20,000	18,400	17,000	14,600	12,800	11,200	10,000	45*	55
	28,000	26,000	24,200	22,400	20,000	17,600	15,600	13,600	46*	56
	70,000	65,000	60,500	57,000	50,000	44,000	39,000	34,000	. . .	57*
Section *B* 70–99 Equiv. ft.	5,000	4,600	4,300	4,000	3,400	3,000	2,700	2,300	41	51
	9,000	8,200	7,600	7,100	6,200	5,400	4,700	4,200	42	52
	11,900	11,000	10,300	9,600	8,400	7,500	6,700	6,000	43	53
	14,800	13,900	13,000	12,200	10,800	9,600	8,500	7,500	44	54
	18,000	16,400	15,200	14,200	12,400	10,800	9,400	8,400	45*	55
	23,900	22,000	20,600	19,200	16,800	15,000	13,400	12,000	46*	56
	59,500	55,000	51,500	48,000	42,000	37,500	33,500	30,000	. . .	57*
Section *C* 100–139 Equiv. ft.	4,100	3,800	3,600	3,400	2,900	2,600	2,300	2,000	41	51
	7,400	6,900	6,400	6,000	5,200	4,600	4,000	3,600	42	52
	10,000	9,300	8,700	8,100	7,100	6,200	5,500	4,800	43	53
	12,500	11,600	10,800	10,100	9,000	8,000	7,200	6,500	44	54
	14,800	13,800	12,800	12,000	10,400	9,200	8,000	7,200	45*	55
	20,000	18,600	17,400	16,200	14,200	12,400	11,000	9,600	46*	56
	50,000	46,500	43,500	40,500	35,500	31,000	27,500	24,000	. . .	57*
Section *D* 140–189 Equiv. ft.	3,700	3,400	3,100	2,900	2,600	2,200	2,000	1,800	41	51
	6,500	6,000	5,600	5,300	4,700	4,100	3,700	3,300	42	52
	8,600	8,000	7,500	7,000	6,100	5,400	4,800	4,300	43	53
	10,900	10,100	9,400	8,800	7,800	6,900	6,100	5,400	44	54
	13,000	12,000	11,200	10,600	9,400	8,200	7,400	6,600	45*	55
	17,200	16,000	15,000	14,000	12,200	10,800	9,600	8,600	46*	56
	43,000	40,000	37,500	35,000	30,500	27,000	24,000	21,500	. . .	57*
Section *E* 190–250 Equiv. ft.	3,200	2,900	2,700	2,500	2,200	1,900	1,700	1,500	41	51
	5,500	5,200	4,800	4,400	3,900	3,400	3,100	2,800	42	52
	7,300	6,800	6,300	5,900	5,100	4,500	3,900	3,500	43	53
	9,100	8,500	7,900	7,500	6,600	5,900	5,200	4,700	44	54
	11,000	10,400	9,600	8,800	7,800	6,800	6,200	5,600	45*	55
	14,600	13,600	12,600	11,800	10,200	9,000	7,800	7,000	46*	56
	36,500	34,000	31,500	29,500	25,500	22,500	19,500	17,500	. . .	57*
For insulated ducts	Col. *a* 1–8	Col. *b* 9–14	Col. *c* 15–20	Col. *d* 21–27	Col. *e* 28–42	Col. *f* 43–54	Col. *g* 55–70	Use these column headings for ducts that are completely insulated with ½-in. thick insulation from bonnet to boot		

* Use these items only when the building construction, or capacity requirements, necessitates the use of two adjoining stacks or floor registers.

are shown column headings to be used in case the ducts have ½-in. insulation. Occasionally it may be necessary to install warm-air registers on cold outside walls. Such ducts, boots, and heads must be protected with at least ½-in. insulation.

TABLE 108.—COMBINATIONS OF PARTS FOR WARM-AIR DUCTS

Combination number	Stack size, in.	Branch pipe size, in.		Register size, in.				Required increase of width of trunk duct, in.
		Round	Rectangular	Baseboard* or low side wall	High side wall†		Floor‡ registers	
					For throws less than 13 ft.	For throws more than 13 ft.		
Col. 1	Col. 2	Col. 3	Col. 4	Col. 5	Col. 6	Col. 7	Col. 8	Col. 9
41	10 × 3¼	6	4 × 8	10 × 6	10 × 6	10 × 4	8 × 10	1
42	10 × 3¼	6	4 × 8	10 × 6	10 × 6	10 × 4	8 × 10	2
43	12 × 3¼	7	5 × 8	12 × 6	12 × 6	12 × 4	9 × 12	3
44	14 × 3¼	8	6 × 8	14 × 6	14 × 6	14 × 4	9 × 12 or longer	4
45	10 × 3¼ (2 stacks)	9	8 × 8	(2) 10 × 6 or (1) 24 × 6	(2) 10 × 6 or (1) 24 × 6	(2) 10 × 4 or (1) 24 × 4	10 × 12	6
46	12 × 3¼ (2 stacks)	10	10 × 8	(2) 12 × 6 or (1) 30 × 6	(2) 12 × 6 or (1) 30 × 6	(2) 12 × 4 or (1) 30 × 4	12 × 14	7

*Recommended type of baseboard and low side-wall registers: Vertical bars with adjustable deflection, or fixed vertical bars with deflections to right and left not exceeding about 22 deg. For low side-wall location, the deflection for horizontal multiple-valve registers should not exceed 22 deg. For baseboard locations, the deflection for horizontal multiple-valve registers should not exceed about 10 deg.

† Recommended type of high side-wall registers: Horizontal valves, in back or in front, to give downward deflections not to exceed from 15 to 22 deg. For deflection angles exceeding 22 deg., note the equivalent lengths given in Table 105.

‡ Use the items in column 8 only when the building construction or capacity requirements necessitate the use of floor registers. The sizes listed for floor registers correspond to the standard sizes for gravity warm-air furnace systems, except for the sizes of the floor box collars. The use of standard blind boxes is suggested.

In the authors' judgment there are many cases where not only should a separate run be installed to rooms over garages and others similarly exposed but such a supply duct should constitute a separate zone, with its own thermostat. The heating requirements of rooms exposed on three sides plus roof and floor are quite different from those of the main parts of the house. The rate at which such rooms cool off between the cycles of operation of the fuel burner is much faster than for the remainder of the house, and a single thermostat does not provide satisfactory control. Neglect of this consideration results in many unsuccessful installations. When there are two or more zones, it is highly desirable to separate the return-air system into corresponding zones, each with its own damper, automatically controlled. This guards against the circulation of unheated air through the rooms whose zone supply damper is closed.

387. Selecting Return-air Combinations.—The method of procedure is substantially comparable to that used in selecting warm-air combinations.

1. Using the bottom of the form outlined in Table 104, indicate on line 13 the rooms to be furnished with return-air openings. In cases where the return serves one or more adjoining rooms, the sum of the respective heat

TABLE 109.—COMBINATION OF PARTS FOR RETURN-AIR DUCTS

Combination number	Return-air intake size, in.		Riser size, in., where stack is used in stud space	Branch pipe size, in.		When joist lining is used,* number of joist spaces lined and minimum depth of space required	Required increase in width of trunk duct (for 8-in. depth of duct), in.
	Baseboard	Floor†		Round	Rectangular		
Col. 1	Col. 2	Col. 3	Col. 4	Col. 5	Col. 6	Col. 7	Col. 8
51	10 × 6	6 × 10 or 4 × 14	10 × 3¼‡	6	4 × 8	1 of 3″	1
52	10 × 6	6 × 10 or 4 × 14	10 × 3¼‡	6	4 × 8	1 of 3″	2
53	12 × 6	6 × 12 or 6 × 14	12 × 3¼§	7	5 × 8	1 of 4″	3
54	14 × 6	6 × 14	14 × 3¼§	8	6 × 8	1 of 5″	4
55	24 × 6 or 30 × 6	6 × 30	Two stacks each 10 × 3¼‡	9	8 × 8	1 of 6″ or 2 of 3″	6
56	30 × 6	6 × 30	Two stacks 12 × 3¼§	10	10 × 8	1 of 7″ or 2 of 4″	7
57	. . . . . .	8 × 30	. . . .	12	15 × 8	1 of 9″ or 2 of 5″	11

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* Based on 14-in. space between joists. Use full depth of joist except when joist depth is less than minimum depth required, in which case a drop pan must be used. This condition may occur when two or more return ducts are connected to the same joist space.

† Use these items only when building construction or capacity requirements necessitate the use of floor intakes. The sizes listed correspond to standard sizes for gravity installations, except floor box collars. The use of standard blind boxes is suggested.

‡ If it is desired to use 14- × 3⅝-in. stud space, it makes no difference whether this space has protruding keys.

§ If it is desired to use 14- × 3⅝-in. stud space, the plaster base must be smooth, without any protruding plaster keys to interfere with the flow of air.

losses is tabulated on line 16. The columns for those rooms that do not have return intakes are left blank.

2. Enter actual length in feet of return duct from boot to plenum on line 14. The sum of the equivalent length of all fittings and intake appearing in each run is listed on line 15. This procedure is exactly the same as that used in selecting warm-air ducts.

3. With the above data select the proper return-air *combination number* from Table 106 or 107 and enter on line 17.

4. From Table 109, tabulate for the proper combination number the stack size on line 18, the branch duct size on line 19, number of joist spaces required and depth on line 20, and the intake or return grille size on line 21.

Return air should be brought back from as many rooms as possible, with the exception of bathrooms, closets, toilets, and garages. Gravity vents are often installed in bathrooms and toilets to dispel objectionable odors. The bathroom return may be vented into the furnace room, with the thought that the air requirements for the combustion device may be partially taken care of in this way.

388. Sizing Trunk Duct.—Basically the procedure of selecting duct sizes is analogous to the methods used in larger systems, as outlined in Chap. XVII. However, the Code attempts to simplify this procedure for residential design by introducing tables that take the friction loss of the system into account. The following steps outline sizing procedure that is applicable to both warm-air and return-air systems.

1. Consider as the beginning of the trunk duct that section of the trunk line which is the greatest distance from the furnace bonnet. Illustrated in Fig. 318 is an example of the method used for duct sizing for the residence shown in Figs. 314 and 315.

2. Determine the equivalent rectangular duct size of the farthest run from column 4 in Table 108 by using the proper combination number.

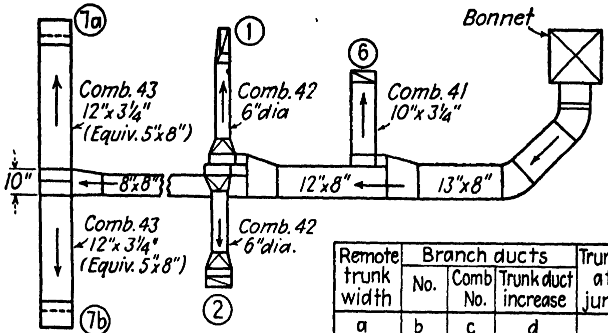
3. From the last columns of Tables 108 and 109, determine the increases to the width of the trunk duct required for each of the branch ducts. The only difference between combination numbers 41 and 42 (or 51 and 52) is in the required increase to the width of the trunk duct.

4. Where the first branch duct joins the trunk duct, add (*a*) the equivalent width of trunk and (*b*) the trunk-duct increase of the branch duct. The sum is the required width of the trunk duct up to the next junction. The depth is generally considered as 8 in.

5. For each of the succeeding junctions, closer to the furnace bonnet, add (*a*) the width of the more remote trunk and (*b*) the trunk-duct increases required for each branch.

6. Where two or more trunk lines join to form a main trunk, each of the smaller trunks should be independently sized up to the junction. The shorter of the two smaller trunks is treated as a branch duct, and the procedure is as previously outlined.

7. In case it is necessary to change the depth of the trunk duct at any point in order to take care of installation limitations, the equivalent duct sizes listed in Table 81, Chap. XVII may be used.



Remote trunk width	Branch ducts			Trunk width after junction
	No.	Comb No.	Trunk duct increase	
a	b	c	d	e
5"	7b	43	3"	8"
8"	1&2	42&42	2"+2"	12"
12"	6	41	1"	13"

FIG. 318.—Example of trunk-duct sizing.

389. Controls.—The Code contains the following requirements for controls:

Blower Controls.—An adjustable automatic fan or blower switch shall be placed in the bonnet of the furnace to start and stop the blower at predetermined air temperatures. The location of the switch, and the adjustment of the setting shall be made according to the recommendations in the Code, unless manufacturer's instructions specify otherwise.

Summer Switch.—Unless automatically controlled, a manual switch for summer operation may be placed in the hallway or other accessible location, but must not be placed where it is liable to be turned by mistake. This switch shall be wired so that it operates the blower independently of the bonnet control.

Limit Control.—A limit control shall be used in conjunction with any automatic temperature control, to prevent overheating of the heater unit. This control shall be so constructed that the temperature of the air in the bonnet cannot exceed about 200°. The usual setting is about 175°.

Hand-fired Coal Furnace.—In addition to the blower switch and limit control, a room thermostat shall be used which operates a damper motor. Damper motors that have current failure protection are preferred types.

Stoker-fired Coal Furnace.—In addition to the blower switch and the limit control, the controls for a stoker-fired furnace shall consist of a room thermostat, a relay for operating the stoker motor, a barometric draft regulator, and some form of hold-fire control.

Oil-fired Furnace.—In addition to the blower switch and limit control, the controls for an oil-fired furnace shall consist of a room thermostat and an approved ignition control.

Gas-fired Furnace.—In addition to the blower switch and limit control, the controls for a gas-fired furnace shall consist of a room thermostat and approved gas valve and pilot safety control.

Zone Controls.—Zone control equipment consists of two or more room thermostats, operating dampers in warm-air ducts leading to various portions, or zones, of the building. Manufacturer's instructions shall be followed in the installation of zone control equipment.

Electric Circuit for Heating Equipment Only.—The owner shall furnish a separate electrical circuit of proper voltage and capacity to be run from the house service to a separate panel. This panel shall be for heating equipment only.

390. Construction Details.—The Code contains the following suggestions and requirements as to details of construction, many of which are quite important:

Wall Stacks.—(Up to and including 14 in. in width.) Wall stacks shall be constructed of galvanized iron sheets, 30 gage or heavier. If slip joints are used they shall be stripped with asbestos paper. Where stack heads, boots, or other fittings, either for warm air or return air, go through first floor, all openings around such fittings shall be filled with asbestos fiber or other noncombustible insulating material to make this opening dust tight. (See requirements of Fire Underwriters.)

Headers.—Tight headers must be installed above all openings for stackheads and return air intakes. (See requirements of Fire Underwriters.)

Round Pipe.—Round pipe shall be constructed of galvanized iron: Up to and including 12 in., not lighter than 30 gage, up to and including 24 in., not lighter than 26 gage. If slip joints are used, joints shall be stripped with asbestos paper.

Rectangular Duct.—Rectangular ducts shall be constructed of galvanized iron: Up to and including 14 in. wide, not lighter than 28 gage; to 36 in. wide, not lighter than 26 gage; wider than 36 in., not lighter than 24 gage. All duct sections 24 in. or wider, and over 48 in. in length, shall be cross broken on top and bottom and shall have standing seams or angle-iron braces. All joints shall be *S* and drive strip, or locked.

No warm-air duct, round or rectangular, shall come in contact with masonry walls. Insulate around all warm-air ducts through masonry walls with not less than ½ in. of insulation. It is not the purpose of this section to eliminate the use of other satisfactory noncombustible materials in the construction of ducts.

Return-air Liners.—Bottoms of all return-air joist spaces shall be lined with smooth airtight material. All cross bracing should be removed and joists properly reinforced.

Insulation of Ducts.—All exposed warm-air ducts passing through an unheated room shall be insulated with at least ½ in. thickness of insulation. Ducts located in cold attic spaces or cold exposed basement spaces shall be insulated with at least 1 in. thickness of insulation.

Duct Supports.—All ducts shall be securely suspended from adjacent building members. No nails shall be driven through duct or stack walls, and no unnecessary holes shall be cut therein.

Volume Dampers.—Volume dampers of locking type shall be placed in each warm-air run. These may be placed near the main trunk or in the stackhead. Return-air ducts and outside air-inlet ducts shall be similarly equipped with volume dampers.

This concludes the abstract of the association Code. It has been dealt with in detail because of its extreme importance in this field of residence air conditioning, and the reader expecting to be active in the field is advised to keep in touch with current revisions.

391. Application of Summer Cooling.—The mechanical warm-air system, if designed in accordance with the Code method, is usually suitable, without modification, for summer cooling. In an important installation, however, it is desirable to make a complete cooling-load calculation and check the air quantities, to be sure that adequate cooling will be obtained in all the rooms. The cooling requirements of all the rooms do not necessarily bear the same relation to the respective heating requirements, and some seasonable adjustments of air quantities may be necessary. However, since many of the rooms in a residence are interconnected and since the permissible range of temperature in summer is quite large, the occupants do not usually object to a reasonable temperature difference between rooms. The upstairs rooms particularly are apt to have higher temperatures in summer.

It is important that a residence be equipped with awnings if it is to be artificially cooled. They will reduce the size of the compressor unit and the cost of operation and will produce more comfortable conditions.

It is not usually necessary to introduce any outdoor air in residence cooling, unless there are more than the usual number of occupants. Natural infiltration usually supplies the needed air renewal.

Methods of cooling and dehumidifying in residences consist of (1) well or cold city water (below 60°), (2) ice, (3) mechanical refrigeration (usually with a direct-expansion coil), and (4) chemical dehydration. The first three methods simply call for an extended-surface coil in the air circuit. If the furnace unit is so designed, as in Fig. 311, the coil is readily added; if not, it can be added in a separate chamber, as shown in Fig. 319. The chemical dehydration system is a self-contained unit, one form of which was shown in Fig. 286, Chap. XVIII.

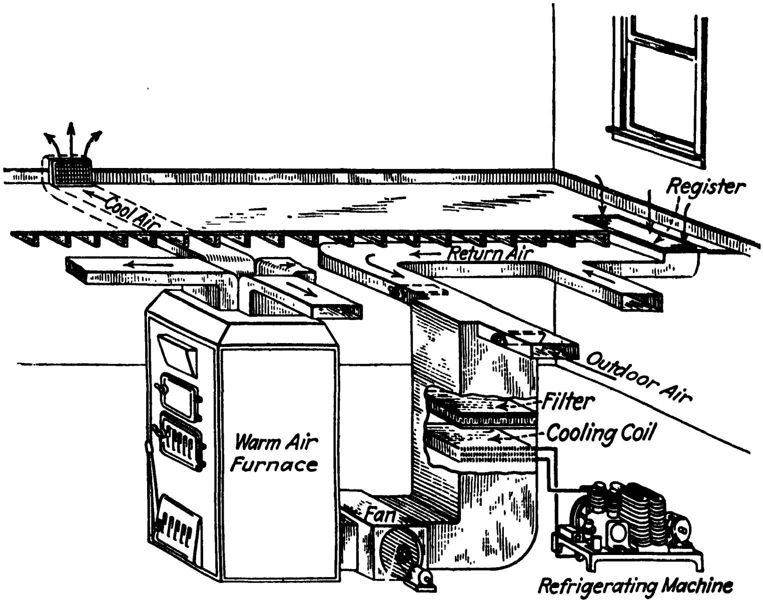


FIG. 319.—Application of mechanical refrigeration to a warm-air furnace system.

There is much difference, in practice, in the size of compressor unit selected by different engineers for comparable conditions. For an eight-room house, near Detroit, of compact design and equipped with awnings, a 2½-hp. compressor unit proved to be just sufficient to hold the maximum temperature (downstairs) to about 80°.[1] The research residence at Urbana, a house of approximately the same size but not insulated, requires about 3 hp.[2] On the other hand, much smaller capacities have been used with considerable benefit, but the temperature may rise to uncomfortable levels for a few hours or days during the season.

TABLE 110.—OPERATING DATA FOR A RESIDENCE NEAR DETROIT USING ICE AND MECHANICAL REFRIGERATION[1]

Year	1932	1933	1934	1935	1936	1937
Cooling medium	Ice	Ice	2-ton ref. mach.	2-ton ref. mach.	2-ton ref. mach.	2-ton ref. mach.
Total degree-hours above 85° during summer	529	1,408	1,345	540	1,847	507
Days of artificial cooling	22	22	23	14	18	12
Hours of artificial cooling	135	134	214	138	245	122
Hours of fan use only						
Outdoor air cooling	57	120	6	57	11	
Recirculation	......	....	114	23	115	
Total hr. fan use	192	254	334	218	371	210
Average cooling-hr. per day	6.1	6.1	9.3	9.9	13.6	10.1
Electrical energy used:						
Ice-water pump and fan, kw.-hr.	94	142				
Compressor, kw.-hr	......	....	437	285	492	244
Fan, kw.-hr (summer only)	......	....	172	114	181	102
Total energy per summer, kw.-hr.	94	142	609	399	673	346
Condensing water for compressor						
Average, g.p.m.	......	....	2.1	2.1	2.1	2.1
Total for season, gal.	......	....	26,700	14,790	30,200	15,100
Cost of cooling for season						
Ice at $5 per ton	$45.60	$30.60				
Electricity at 2¼ cents per kw.-hr.	2.12	3.20	$13.71	$ 9.00	$15.15	$ 7.76
Condensing water at 20 cents per 1,000 gal	......	....	5.34	2.96	6.04	3.02
Total operating cost	47.72	33.80	19.05	11.96	21.19	10.78
Average daily cost of artificial cooling	2.16	1.54	0.83	0.85	1.17	0.90
Average hourly cost of artificial cooling	0.35	0.25	0.089	0.087	0.086	0.088

[1] The residence is a two-story eight-room building of frame construction 46 by 26 ft. in plan. Total space cooled 17,200 cu. ft. All openings weather-stripped. Walls built up of shingles, three-ply paper, sheathing, studding, lath, and plaster, with flexible insulation between the studding. Attic floor insulated. Shutters or awnings on all windows exposed to the sun. Winter heat loss 93,000 B.t.u. per hr. (calculated) for 70° inside and 0° outside. For further details see the paper in ref. 1 p. 598.

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[1] HELMRICH and WALKER, "Summer Cooling Operating Results in a Detroit Residence," *Trans. A.S.H. & V.E.*, 1933.

[2] KRATZ and KONZO, "Study of Summer Cooling in the Research Residence at the University of Illinois," *Trans. A.S.H. & V.E.*, 1933.

Table 110 shows the yearly costs of operating the cooling system of the residence near Detroit, which first used ice and later a mechanical compressor unit. The local costs for ice and electricity are shown, but the relation between them will differ in different cities, and the figures therefore give a general comparison between the two methods only as to the quantities used.

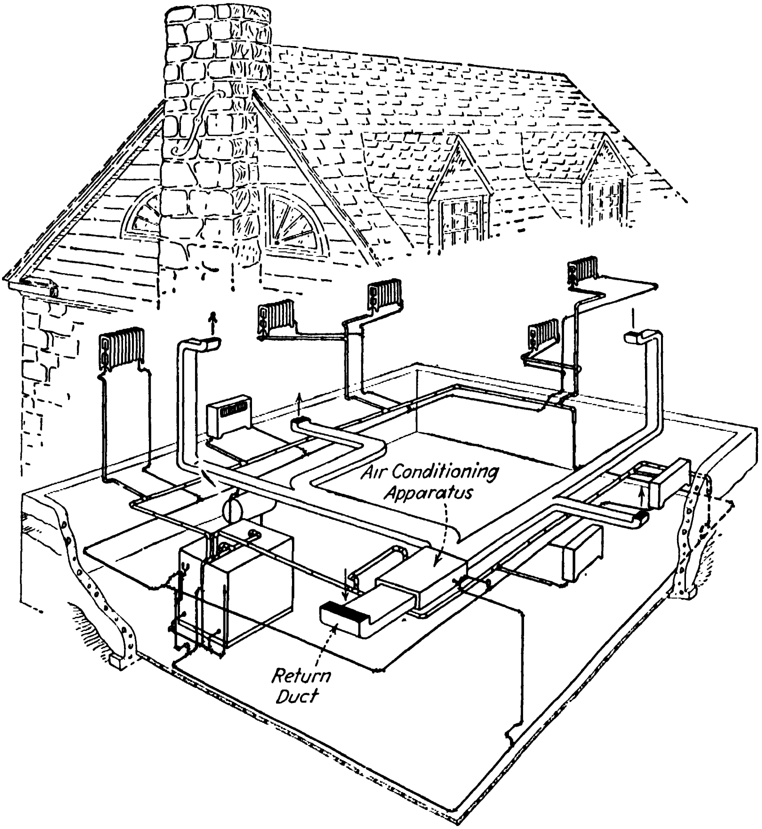

FIG. 320.—Air conditioning with radiator heating. (*American Radiator Company.*)

In general it may be said that the widespread use of artificial cooling in residences requires the development of a low-cost, simple cooling apparatus. The cost of installation of a compressor and coil now ranges from $500 to $700 and is a more formidable factor in most cases than the operating cost.

392. Air Conditioning with Radiator Heating.—Manufacturers of radiators offer supplementary air-conditioning apparatus for use with radiator heating. The system is quite analogous to the split system of ventilation or air conditioning (Chap. XV.) It consists of a central fan with filter, humidifier, tempering heater, and, if desired, a cooling coil. Figure 320 shows the method of application. It is usually designed to serve only the main rooms of the residence. The methods of design are similar to those already outlined.

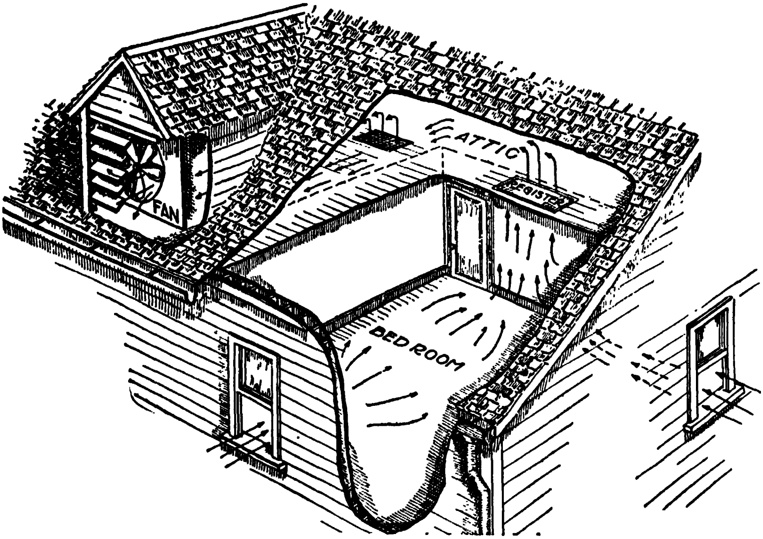

Fig. 321.—Attic-fan installation.

This type of air conditioning has not been widely applied but is a practicable method of treating a radiator-heated house.

393. Units Applied to Residences.—Air-conditioning units, which will be discussed in the next chapter, offer a practicable method of cooling single rooms or a few rooms closely interconnected. Since the units are completely self-contained, not much application engineering is required. The sizes used for most rooms are ½ to ¾ ton of refrigeration.

394. The Attic Fan.—The use of an attic fan for drawing outdoor air through the house is an important and valuable method of cooling residences. It is based on the fact that in most parts of the country the outdoor temperature falls, in the evening, into the region of comfort, whereas the inside is very warm from the heat absorbed by the structure during the day. If this outdoor air can be drawn into the house in sufficient quantities, it can do a fairly good job of cooling, partly by virtue of its lower temperature and partly because of the air movement created. The attic fan, as illustrated in Fig. 321, simply draws into the house, and especially into the bedrooms, a copious quantity of outdoor air. The effect is quite like a strong outdoor breeze. For best results a capacity of 20 to 30 air renewals per hour for the second-floor bedroom should be allowed. This large supply of air, in addition to lowering the temperature, creates a noticeable air movement. Figure 322 shows the temperature reduction obtainable.

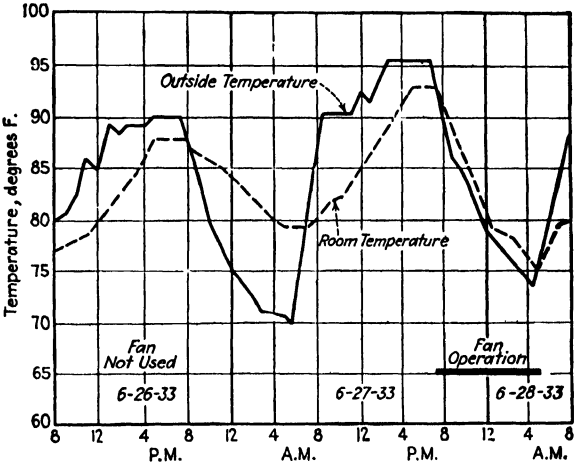

Fig. 322.—Cooling effect with attic fan.

In the Southern states where the outdoor temperature does not fall much at night, a considerably greater fan capacity—up to 60 air renewals—has been found necessary. The fan should be selected for quietness and mounted on sound-insulating supports.

For a low-cost method (as to both installation and operation) of cooling a residence at night the attic fan is to be recommended.

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"Air Conditioning for Houses in California," by Gordon and Perry, *Univ. Calif. Circ.* 351, May, 1942.

"Winter Air Conditioning," by S. Konzo.

Problem

1. Using the design data of Prob. 1, Chap. IV, determine for a mechanical warm-air furnace system the required warm-air combination stack sizes, branch pipe rectangular dimensions, and baseboard register sizes for all the rooms listed. The following table gives data pertaining to the actual duct length of bonnet to boot and equivalent fitting lengths:

Room number	Use	Length warm-air supply duct, ft.	Length equivalent fittings, ft.
1	Living Room	15	135
2	Dining Room	13	120
3	Kitchen	6	95
4	Hall	6	95
5	Bedroom	13	130
6	Bedroom	6	90
7	Bedroom	13	140
8	Bath	6	90

CHAPTER XXII

UNIT AIR CONDITIONERS

395. Types of Units.—Units for air conditioning or for heating or cooling should be accurately termed in accordance with the exact functions they perform. The importance of exact descriptive terminology is due to the misunderstanding on the part of the public and in some cases to misrepresentation as to what a particular piece of apparatus accomplishes.

Cooling units, which with the addition of filters and controls become air-conditioning units, are of two general types: (1) those supplied with chilled water or refrigerant and (2) those which are self-contained and include a mechanical refrigeration compressor.

There are different forms of both of the foregoing types, to meet installation requirements. There are **overhead units** designed to hang from the ceiling. **Room units** are enclosed in a finished cabinet of refined design. **Floor units** stand on the floor, as differentiated from overhead units.

396. Chilled-water Units.—This type of unit is desirable where it is convenient to pipe chilled water to the units. As compared with the self-contained unit it is much less costly, takes less space, and requires comparatively little maintenance.

The chilled water may be obtained from a well, from the city mains (if below 60°), or from a refrigerating machine. It is seldom appropriate to install a refrigerating machine to chill water for only a few conditioning units, but occasionly conditioners are installed in individual rooms when there is already a central system using chilled water in another part of the building.

Figure 323 shows a room unit using chilled water. It requires three pipe connections—water supply and return, and a waste line from the drip pan. This is a cooling unit only and has no filter, but filter sections can be added in this, as in most designs.

There is naturally an appreciable amount of condensation dripping from the coil surface, and it is necessary to have a drip pan, which must be drained to the sewer. Local plumbing regulations govern the piping arrangements.

Temperature control of the cooling function is readily accomplished by means of a thermostat actuating a solenoid valve in the chilled-water supply pipe.

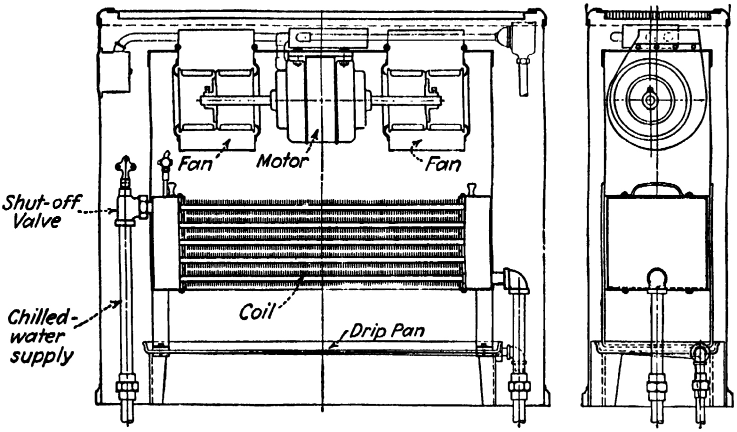

Fig. 323.—Cooling unit using chilled water. (*American Blower Corporation.*)

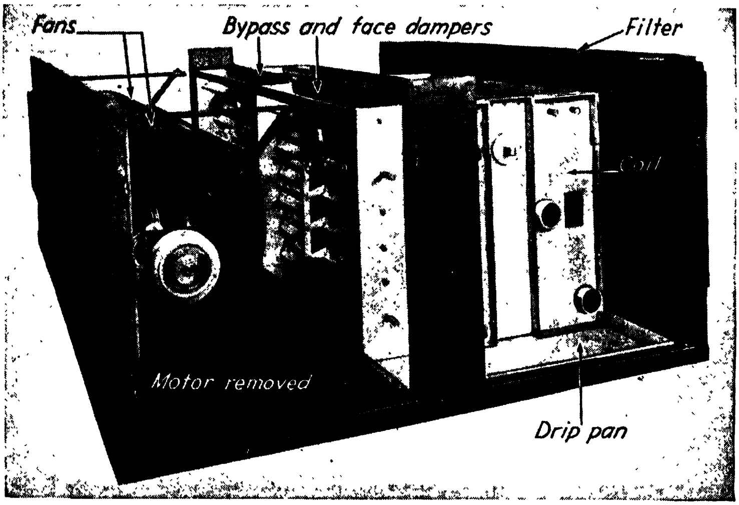

Fig. 324.—Unit conditioner of the suspended type (casing removed). (*American Blower Corporation.*)

Units of this type may be designed for cooling or for both cooling and winter ventilation or heating. When used for cooling, in rooms where there are many people or where there is smoking, it is important that some outside air be introduced; otherwise odors will undoubtedly build up. Figure 324 shows the interior of a unit conditioner of the suspended type.

Air-induction Units.—One variation of the above type of unit is a combination system, which makes use of the central air-conditioning plant discussed in Chaps. XV and XVIII and a unit system of distribution. Generally a small amount of air, including air for ventilation requirements, is conditioned by a large central plant and then transmitted to each individual space through a system of ducts or tubing.

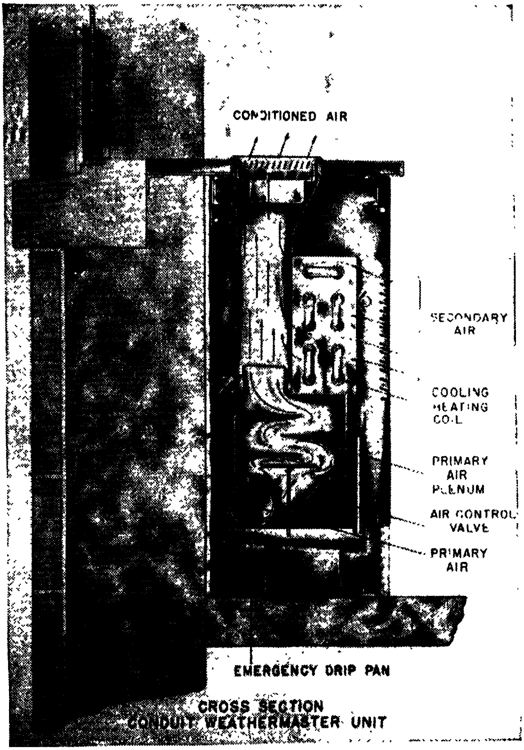

Fig. 325.—Air-induction unit. (*Carrier Corporation.*)

This primary air is delivered to each room through an *injector* type of unit as shown in Fig. 325. The unit usually consists of a booster heating or cooling coil, or both, and some type of nozzle for introducing air to the rooms at high enough velocities to *induce* a large secondary air circulation.

The room unit requires no fans nor individual outdoor air connection. However, a high-pressure-type fan is generally used to convey the primary dehumidified air through small-sized ducts or tubing, which is customarily concealed in furred spaces along with the chilled or hot-water piping.

Refrigerant-supplied Units.—It is often more convenient to install a mechanical refrigerating unit to supply one or several conditioners with refrigerant, which is expanded directly in the coils of the units. Units for this purpose are almost identical with chilled-water units except that the coil is designed to use the refrigerant, and certain features such as the refrigerant expansion valve are required.

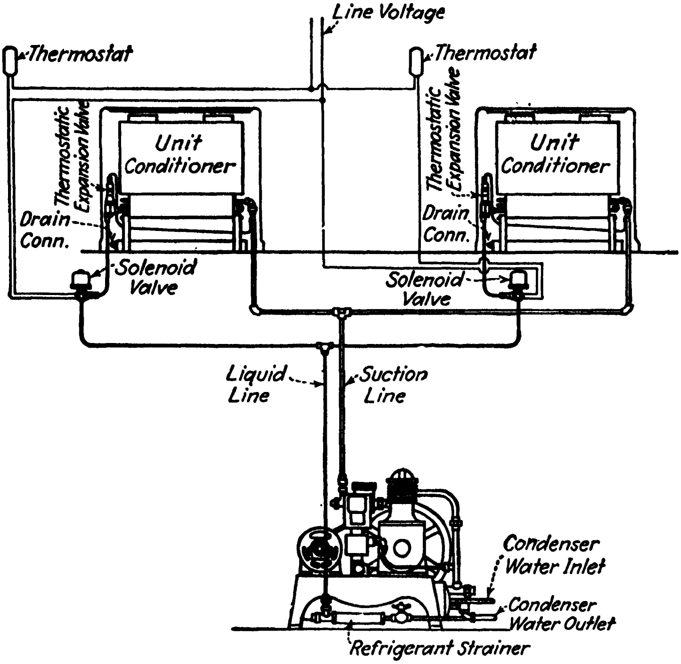

FIG. 326.—Unit conditioners supplied by a compressor. (*American Blower Corporation.*)

Figure 326 shows a piping and wiring diagram for two units, with separate temperature control. Local regulations sometimes call for an outside dump valve for the refrigerant, to be used in case of fire.

Self-contained Units.—Where it is not convenient to pipe refrigerant or chilled water to the units, the self-contained type may be used. This type has its own compressor unit incorporated within the casing. The unit is heavier, bulkier, and costlier than the chilled-water type and requires more maintenance.

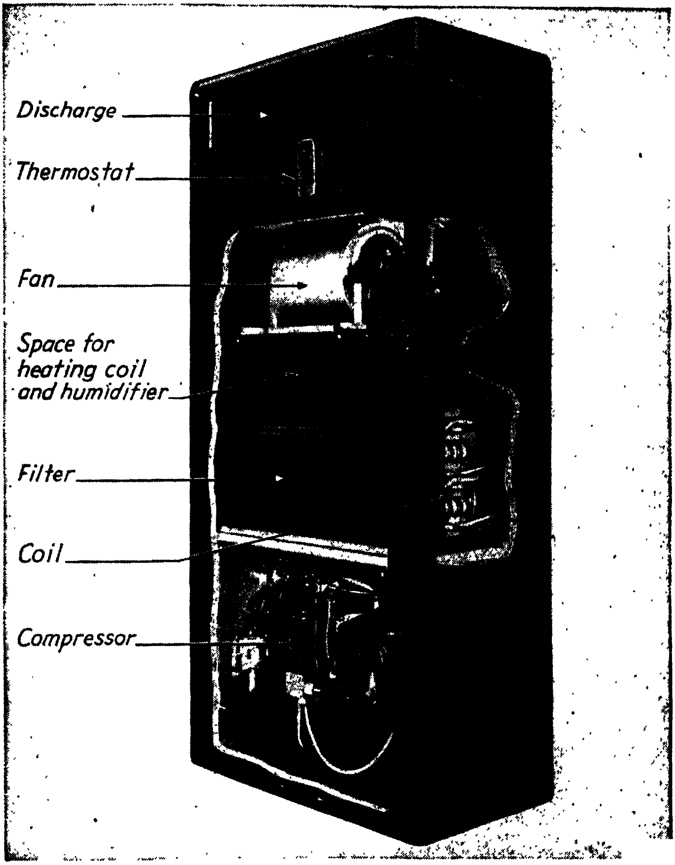

FIG. 327.—Self-contained store-type unit. (*Kelvinator Corporation.*)

Figure 327 shows such a unit of about 30,000 B.t.u. per hr. capacity. It is of the type well adapted to small retail stores and is often termed a store-type unit. A certain amount of ductwork can be connected to the discharge for proper distribution of the air, as indicated in Fig. 328 but it is important to make sure that the fans are of proper design to handle such additional resistance.

It is evident that such a unit, with ductwork, is not basically different from a built-up central-fan system. Its advantage is that it requires little field work, is more compact, and is usually less costly then the specially assembled apparatus.

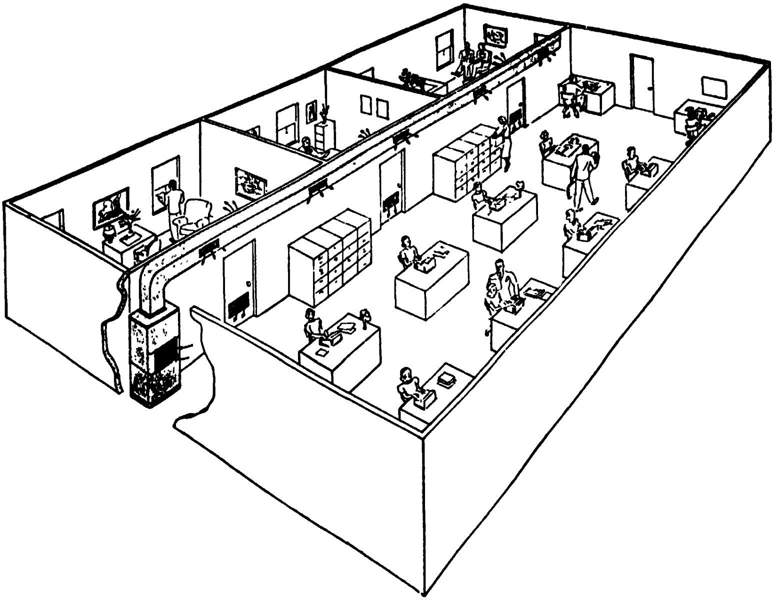

Fig. 328.—Store-type unit serving a suite of offices. (*Kelvinator Corporation.*)

397. Air-cooled Condensers.—The unit of Fig. 327 requires piping connections for condenser water supply and discharge and for the drip pan. It is sometimes inconvenient and costly to install such connections. As an alternative, the condenser may be air-cooled and so arranged that the drainage from the coil is allowed to drip over the condenser and is evaporated into the cooling air. This type of unit of course requires an outdoor air connection of generous size and is not always suitable because of the difficulty of making such a connection. Figure 329 shows such a unit, of the floor type. It has two fans—one for the room air and one for the condenser air, the latter driven from the compressor motor.

398. Application.—The engineering involved in an air-conditioning system making use of units consists of (*a*) choosing the proper type of units, (*b*) selecting the proper sizes, and (*c*) arranging them in the proper locations.

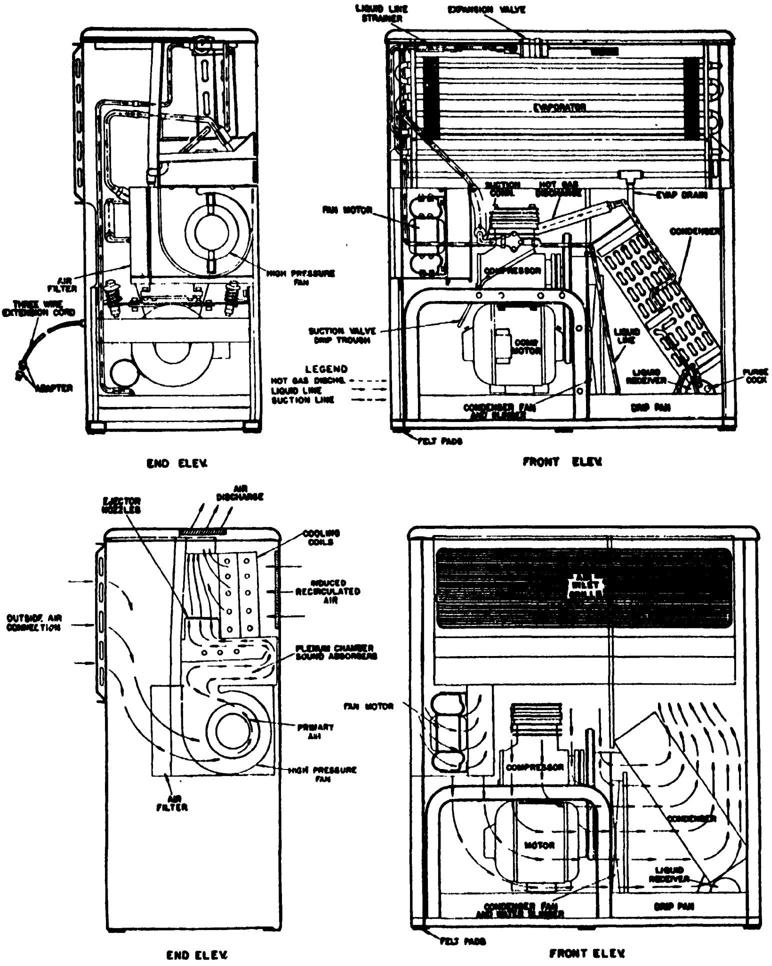

Fig. 329.—Floor-type, self-contained unit with air-cooled condenser. Lower view shows air circuits. (*Carrier Corporation.*)

The basis of the size selection is of course a load calculation, which is exactly like that required for a central system, although in a single room it need not usually be carried through with great accuracy, because the steps between unit sizes are relatively large.

The mistake should not be made of putting in a much-undersized unit in the hope of getting a little cooling effect. The result of this has usually been that the occupants are less comfortable with the windows closed and the unit operating than they are with the windows open and the breeze blowing in. Under the latter conditions the air-conditioning unit is of course of no value whatever. Nor is it often practicable to cool an adjacent room through an open door—a mistake not infrequently made.

Because of the outdoor air requirements, either for ventilation or for the condenser, units are most often placed under windows. They discharge upward in many cases, and the air circulation thus obtained is usually satisfactory and free from serious drafts.

Owing to lack of careful design, the use of units of the floor type for winter ventilation has not been altogether satisfactory in the past, but they have been somewhat improved in that respect. Connections to the window opening must be absolutely airtight to avoid drafty leaks. The heating coil must be so arranged as to give a uniform temperature over the face of the outlet grille.

399. Ratings.—The following approximate range of rated[1] capacities is available for the different types of units. It varies somewhat with different manufacturers:

	Capacity, Tons of Refrigeration
Suspended units	5 to 40
Floor (cabinet-type) units (chilled water or refrigerant)	0.5 to 5
Self-contained units:	
Floor type	0.5 to 0.75
Store type	3 to 5

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[1] See "Standard Method of Rating and Testing Air Conditioning Equipment," A.S.R.E. *Circ.* 13-42, and "Standard Methods of Rating and Testing Self-contained Air Conditioning Units for Comfort Cooling," A.S.R.E. *Circ.* 16.

CHAPTER XXIII

INDUSTRIAL AIR CONDITIONING

400. Applications in Various Industries.—The use of air conditioning is of advantage in many kinds of industries, either because of the greater comfort and efficiency of the workers or because of the better grade of product obtained. In fact, these conditions often go hand in hand, the proper atmosphere from a manufacturing standpoint being in many cases also the more desirable from the standpoint of comfort and health.

The following paragraphs quoted from a paper presented by Willis H. Carrier summarize the present applications of industrial air conditioning.

> It has long been recognized that relative humidity is an important factor in the manufacture and processing of certain hygroscopic materials such as textiles. Since the normal relative humidity in textile factories is nearly always lower than that desired, various artificial means have for many years been provided to increase the humidity.
>
> On the other hand, there are industries which require a definite and unvaried humidity so that at some periods the normal quantity of moisture in the air must be increased and at other times lowered. Among such industries may be mentioned the manufacture of confectionery, the processing and weaving of artificial silk, and the printing and lithographing industry.
>
> Other industries require not only a constant relative humidity but also a fairly uniform temperature. An example of this is seen in the modern automatic wrapping machines, used for wrapping chewing gum, food products, confectionery, and machine-made cigarettes, which require exact conditions of heat and moisture in order to function satisfactorily without frequent adjustments.
>
> In the manufacturing and processing of most hygroscopic materials there are usually one or more stages in the process in which moisture has to be removed from the material. In all such products which are not in themselves soluble in water, this is usually accomplished by air drying.
>
> In order to avoid injury to the products which require temperature and humidity control, the rate of moisture removal must be controlled

with accuracy in certain stages of the process, while at the end of the process of drying it may be extremely important to control with accuracy the final moisture content of the material. This is true in the manufacture of the various tobacco products, artificial silk, certain gelatine products including photographic films, lumber, paper, and macaroni. In many of these products it is not only necessary to have this final moisture content to ensure the desirable qualities of the product itself, but to standardize its moisture content from the standpoint of sales value; as for example, in artificial silk, of which hundreds of millions of dollars' worth is manufactured annually, 1 per cent variation in moisture content would mean millions of dollars variation in the sales price of the product, although not in its intrinsic value. The same is true to a lesser extent in cotton yarns, with which an increased moisture content is not only desirable from the sales standpoint but from the manufacturing standpoint as well. Wheat flour is another product that requires careful moisture standardization, and considerable attention is being given to this field at the present time.

Air conditioning is important in certain branches of the chemical industry in controlling the temperature of reaction and facilitating or retarding evaporation.

Table 111 is a list of materials and processes for which air conditioning is desirable. In some of these fields its possibilities are well recognized, but in others its value is only beginning to be appreciated.

401. Hygroscopic Properties of Materials.—Air conditioning in many industries has to do with the control of moisture in manufactured goods. The hygroscopic moisture varies considerably in different materials, but its amount depends upon the *relative humidity* of the surrounding air, and for each condition of relative humidity and temperature there is a certain equilibrium condition of moisture content for any material. The moisture content of hygroscopic materials is called **regain,** a term which is somewhat of a misnomer but which arises from the fact that hygroscopic materials, if completely dried and exposed to moist air, will regain a moisture content, the amount of which depends upon the relative humidity of the surrounding air. A typical regain chart is shown in Fig. 330 for a temperature of 70° for cotton and paper.

If the process is one of drying, the rate at which the moisture is removed must in many instances be retarded in order to avoid casehardening, or surface drying. The principle to be observed

TABLE 111.—MATERIALS AND PROCESSES FOR WHICH AIR CONDITIONING IS DESIRABLE

Product or industry	Process	Air conditioning essential for
Ammunition	Powder and explosives processing	Fixing rate of burning Accurate weighing
	Ballistics, detonators, fuses, igniter packing, pompom mix	Static-electricity and humidity control Precision of parts
	Grinding, priming pressing, solvent reclamation	Static-electricity and humidity control
	Tetryl blending and pressing	Static-electricity control and prevention of explosions
Brewing	Fermentation	Uniformity of quality and flavor Lessens danger of air-borne contamination
Chemicals and pharmaceuticals	Production of capsules, colloids, crystals, serums, and tablets	Control of rates and limits of chemical reactions Quality and purity control
	Penicillin	Control of spore production
Engines and airplanes	Rivet cooling Parts machining Coolant refrigeration Propeller processing Paint application Inspection and assembly of parts	Controlled precision and uniform tolerances Elimination of effects of humidity and temperature variances Quality control
	Drafting rooms	Cooling and dehumidification of room air to improve quality of prints, protect drawings, and increase personal efficiency
Electrical products	Motors and controls: Remote-control equipment	Control of precision and tolerance Dust elimination

TABLE 111.—MATERIALS AND PROCESSES FOR WHICH AIR CONDITIONING IS DESIRABLE.—(*Continued*)

Product or industry	Process	Air conditioning essential for
Electrical products (*cont.*)	Radio equipment Telephone equipment	Prevention of damage by humidity and perspiration action
	Electric cables	Low humidity
Food:		
Bakeries	Flour storage	Mold prevention
	Fermenting	Control of fermentation
	Dough mixing	Dough retarding
	Water cooling	Mix water cooling
	Cream storage	Spoilage prevention
Candy	Coating, dipping, cooling, and packing	Uniformity of product Spoilage prevention
Cereals	Preparation materials, packaging and sealing	Quality and purity control
Macaroni	Curing, fermentation and drying	Control of fermentation and prevention of strains or cracking
Meats	Cutting and packing	Spoilage prevention
Dairy products	Milk and cream aeration	Bacteria control
	Milk and butter storage	Spoilage prevention
Fruits and vegetables	Cold storage	Spoilage prevention Quality and flavor control
Furs	Drying and storage	High quality and spoilage prevention
Glass	Intermediate steps in processing	Uniformity of product Quality of product Production of nonshatterable glass
Instruments: Bombsights Watches Navigation Fire control	Tool processing Parts processing Inspection Assembly	Precision and tolerance control Dust and condensate elimination Production uniformity
Laboratories	Industrial testing Assaying Growth and sprouting Crystallization	Assuring accurate control of temperature and humidity

TABLE 111.—MATERIALS AND PROCESSES FOR WHICH AIR CONDITIONING IS DESIRABLE.—(*Continued*)

Product of industry	Process	Air conditioning essential for
Machine shops	Coolant refrigeration Assembly Inspection	Assuring proper tolerances Providing dust and moisture control
Optical	Lens development Lens assembly Cementing Precision room Experimental room	Precision and tolerance control Mar prevention Condensation and dust control
Petroleum products	Processing Testing Laboratory	Uniform production Octane-requirements control Performance determination
Plastics	Intermediate stages of processing	Reaction control Uniform production
Paints and lacquers	Pigment processing	Control of air-moisture needs, drying rate, etc. Preventing explosions
Paper	Processing Testing Storing	Control of strength and sizing and moisture content
Photographic film	Coating Drying Packing Storage Developing	Uniformity of thickness, pliability, and coating Control of humidity and spoilage prevention Control of bath temperature
Printing	Binding Folding Pressroom Storage of rollers	Increased production Optimum register for multicolor processes
Rubber	Drying and curing Latex processing Roller cooling	Process, schedule, and quantity control

TABLE 111.—MATERIALS AND PROCESSES FOR WHICH AIR CONDITIONING IS DESIRABLE.—(*Concluded*)

Product or industry	Process	Air conditioning essential for
Steel	Blast furnaces Testing Storage	Increased production Rust prevention
Synthetic products: Rubber Nylon and rayon	 Manufacturing Spinning, throwing, and weaving	 Making possible uniform production
Telephone exchanges	Apparatus room Switchboards	Control of static Prevention of sparking Eliminating corrosive action
Textiles	Spinning Weaving Parachute making	Static control Uniformity of production
Tobacco	Manufacture of cigars and cigarettes Softening, curing, stemming, or stripping	Rate of production and appearance Regulating final moisture content

is that the air in contact with the material must be sufficiently moist in itself to keep the moisture from being removed from the external surface of the material faster than it flows from the interior of the piece of material to the surface.

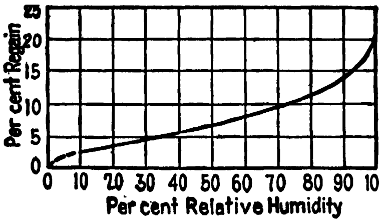

FIG. 330.—Regain chart for sea-island cotton and cellulose paper, at 70°. (*Carrier Corporation.*)

402. Conditions Required for Various Industries.—The temperatures and humidities in Table 112 show what is required for certain important industries. Since this chapter is intended to give only a general picture of the problem of industrial air conditioning, it is not intended that the figures be used in design without verification.

TABLE 112.—TEMPERATURES AND HUMIDITIES REQUIRED FOR VARIOUS OPERATIONS

Operation	Temperature, °F.	Relative humidity, per cent
Bakeries:		
Dough room	80	75 to 80
Cooling room	70	60 to 70
Flour storage	70 to 80	60
Yeast storage	30 to 40	60 to 75
Ceramic factories:		
Refractory drying	110 to 150	50 to 60
Molding room	80	60
Candy factories:		
Chewing-gum rolling room	75	50
Hard-candy manufacturing	70 to 80	35 to 50
Electrical products:		
Paper-insulated cable winding	105	5
Cotton-insulated wire manufacture	60 to 80	60 to 70
Food products:		
Macaroni	70 to 80	40
Cereals	60 to 70	40
Meat hanging	40	80
Paper:		
Manufacture of books, etc.	60 to 80	25 to 50
Printing	75	60 to 75
Photographic film:		
Drying	75 to 80	50
Textiles:		
Cotton carding	75 to 80	50
Cotton spinning	60 to 80	60 to 70
Cotton weaving	70 to 75	70 to 80
Rayon spinning	70	85
Rayon twisting	70	65
Silk spinning and throwing	75 to 80	65 to 70
Silk weaving	75 to 80	60 to 70
Tobacco:		
Cigar and cigarette manufacture	70 to 75	55 to 65

The quantities of air, heat, and moisture that must be handled are next to be ascertained, and the design of the system itself can then proceed.

403. Methods.—In general, the apparatus is similar to that used for comfort air conditioning. For humidifying, direct humidifiers of the atomizing type are often used, and they are located in the processing rooms. The spray washer is used for both humidifying and dehumidifying and is usually preferred to coils for the latter purpose because of the more accurate control obtainable.

Control of humidity is accomplished in either of two ways:

1. *The Dew-point Method.*—This is particularly suitable when the same moisture content in the incoming air is desired without frequent readjustment. It is not very workable when varying quantities of moisture are produced or absorbed in the various rooms but is quite satisfactory and is frequently used where the product absorbs or gives off moisture at a uniform rate. In such cases the dew point at the washer outlet is adjusted to compensate for this absorption or liberation.

2. *Direct relative-humidity control* is preferable if different humidities must be carried in different rooms. It is also the best method if direct-type humidifiers are used. This method, as explained in Chap. XI, uses an element of a hygroscopic material, which actuates suitable compressed-air or electrical devices to open and close valves or dampers.

The methods of control and the ways in which they are applied vary so much with different manufacturing conditions that it is impossible to set down any fixed rules of design.

Reheating is not infrequently used in industrial work where low humidities are required or where temperature and humidity must both be accurately controlled. For low humidities the dehydration method is often appropriate.

404. Comparison with Comfort Air Conditioning.—Industrial air conditioning is based on the same psychrometric principles as comfort air conditioning and makes use of the same general forms of apparatus, but there are important differences in the atmospheric conditions which must be maintained. Different industries require vastly different conditions, some of which are extreme. For example, in the manufacture of electric cables a relative humidity of 5 per cent is required, whereas a cotton-weaving mill requires 70 to 80 per cent. Temperatures also vary widely for different processes.

Although comfort of the workers is not the object sought in industrial-process air conditioning, it is necessary to consider the probable efficiency of the employees and their ability to withstand the proposed conditions. It may be necessary to make a compromise between the two standards. Investigations are being made of the workers' ability to endure high humidity coupled with high temperatures in such places as textile mills.[1]

In some processes accurate control of temperature is important; in others control of humidity is more necessary; and in a few both temperature and humidity must be held within close limits. Where both must be controlled, the apparatus and particularly the control equipment become rather complex as compared with comfort installations.

In comfort work the heat and moisture to be handled are largely dependent on the outside conditions and on the number of occupants. Furthermore, the inside conditions to be maintained usually depend upon the outside conditions. In industrial work the heat and moisture are often produced by the process, and usually the conditions are to be kept constant without regard to the outside conditions.

In attacking an industrial air-conditioning problem, the first step is to make a thorough study of the product, the process of manufacture, and the exact atmospheric conditions required for each operation. In most industries those in charge of production are familiar with the conditions required, but this should not necessarily be assumed in all cases.

405. Methods of Air Distribution.—The primary requirement in the air-distribution system is that the air be distributed with respect to the sources of heat.

One method of distribution is to discharge the air from fan-shaped outlets that spread it horizontally. Extreme quietness of operation is not usually essential, nor is a low rate of air movement always necessary. In fact, in some cases a rapid and general movement of air is particularly desirable. To accomplish it, the air is sometimes introduced through a row of cone-shaped nozzles near the ceiling, which discharge air at a rather high velocity, creating by their aspirating effect a secondary movement of considerable volume.

The so-called "pan system" is also sometimes used, consisting of ceiling inlets with a horizontal plate below, which serves to deflect the air horizontally in all directions. By this method the air may be introduced with a minimum amount of disturbance which is sometimes desirable.

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[1] FLEISHER, STACEY, and HOUGHTEN, "Air Conditioning in Industry," *Trans. A.S.H. & V.E.*, 1939.

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"Air Conditioning in Industry," by M. O. Soroka, *Heating and Ventilating* ("Handling of Foods," November, 1936) ("Manufacture of Matches, Textiles and Tobacco Products," December, 1936) ("Paper Making, Printing, Lithographing and Plastic Manufacturing," February 1937) ("Baking and Fermentation," March, 1937) ("Manufacture of Dyes, Glass, Concrete and Pharmaceuticals," April, 1937) ("Improved Efficiency of Workmen," May, 1937) and ("What Are the Best Conditions for Workers?" June, 1937).

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As we have intense desire for sexual intercourse, O' Lord thou also hast got desire to have creation.

APPENDIX

THEORY OF AIR AND WATER-VAPOR MIXTURES

The presentation in Chap. XIII involving a simple conception of air and water-vapor mixtures (sometimes referred to as moist air) is sufficiently accurate for all practical purposes. However, the recent work of Goff and others has shown that the true laws governing such mixtures are quite complex. Therefore where precise calculations are required, the following factors require consideration, and they should in any case be studied by anyone desiring a complete understanding of the phenomena involved.

The familiar **perfect-gas law** ($PV = WRT$) given in Eq. (3), Chap. XIII, is now recognized according to modern experimental methods to be only approximately correct. Whereas according to Boyle's law the gas isotherms are straight on a PV, P plane, they are not horizontal but inclined downward to the right at relatively low temperatures and upward to the right at high temperatures. This results from the effect of intermolecular forces on the thermal properties of gas at extremely low and high temperatures and is represented as the last term in Eq. (A).

$$PV = WRT - A(T)P \tag{A}$$

Values of $A(T)$, a temperature function called *second virial coefficient,* are known with satisfactory accuracy for both dry air and water vapor. The fact that $A(T)$ is multiplied by P in Eq. (A) indicates that intermolecular forces vanish at zero pressure and infinite volume.

The specific volume of dry air at 64° and 29.921 in. Hg calculated according to Eq. (A) is 13.1977 cu. ft. per lb., and according to Eq. (3), Chap. XIII, is 13.200 cu. ft. per lb., the difference being due to the intermolecular forces.

The above comparison indicates that the error resulting from computing values of specific volume by the two equations is extremely small. Thermodynamically, however, Eq. (3), Chap. XIII, denies the effect of pressure on the thermal properties of a gas, which experiment has shown to be appreciable.

In the past, the thermodynamic properties of moist air have been predicted from available information regarding its two constituents, dry air and water vapor, on the basis of **Dalton's law.** If both dry air and water vapor are regarded as perfect gases, the procedure is consistent. Despite its great advantage, the application of Dalton's law in the case of moist air is inaccurate because neither the dry air nor water vapor can be regarded as a perfect gas. Attempts have been made to improve the accuracy by using the properties of water vapor from steam tables, but this procedure is also thermodynamically inconsistent.

Departures from Dalton's law arise from the fact that the action of intermolecular forces is taking place. Considering water vapor separately, the effect of these forces on its properties has been observed and formulated into steam tables; and similar information regarding dry air is available when the latter is considered separately. Yet when dry air and water vapor are mixed together to form moist air, forces of interaction take place, the effects of which have not until recently been experimentally observed.

The essence of **Dalton's law of partial pressures,** as referred to in Chap. XIII, is that each constituent gas in a mixture such as air and water vapor behaves exactly as if it alone occupied volume V at temperature T of the mixture. Under such conditions the observed pressure P would be the sum of the individual partial pressures of air and water vapor, and the total enthalpy would be the sum of the individual enthalpies. As mentioned above, this is true only if the effect of intermolecular forces is ignored. When the latter is considered, Eq. (A) takes the form of

$$PV = WRT - (A_{aa}x^2 + 2A_{aw}x(1 - x) + A_{ww}(1 - x)^2P \qquad (B)$$

where A_{aa} = second virial coefficient for dry air expressing intermolecular forces between air and air molecules.
A_{ww} = second virial coefficient for water vapor expressing intermolecular forces between water and water molecules.
A_{aw} = interaction coefficient expressing intermolecular forces between air and water molecules.
x = unit weight dry air per unit weight mixture.

Values of A_{aa} and A_{ww} are known, but until recently no reliable information on the interaction coefficient A_{aw} has been available. The substitution of proper coefficients in Eq. (B) and the subsequent determination of the saturation pressure for pure water vapor indicate variations due to departures from Dalton's law of about 0.5 per cent at atmospheric pressure. According to Dalton's law the weight of water vapor per pound dry air at 68° and atmospheric pressure is 0.01468 lb. (Table 66), whereas the correct value, taking into effect the intermolecular forces, is 0.01475 lb. At 100 lb. per sq. in. pressure, the error is nearer 15 per cent for the saturation pressure of water vapor when the correct interaction coefficient is considered.

Relative humidity (Φ) as defined in Art. 221 (Chap. XIII) is the ratio of the actual vapor pressure to the vapor pressure of saturated air at the same temperature. Here again this statement lacks thermodynamic soundness because of departures from Dalton's law, although it possesses practical merit as a useful approximation.

A true measure of the percentage of water vapor in a mixture may be referred to as **per cent saturation** (μ) and is defined as the ratio of the actual weight of water vapor per pound of dry air to weight of water vapor at saturation and at the same temperature and pressure.

Goff has shown that the relation between *per cent saturation* (μ) and relative humidity (Φ) may be translated from the empirical Eq. (C):

$$\mu = \Phi - C\Phi(1 - \Phi) \qquad (C)$$

where C depends on temperature for standard atmospheric pressures as shown in the following table.

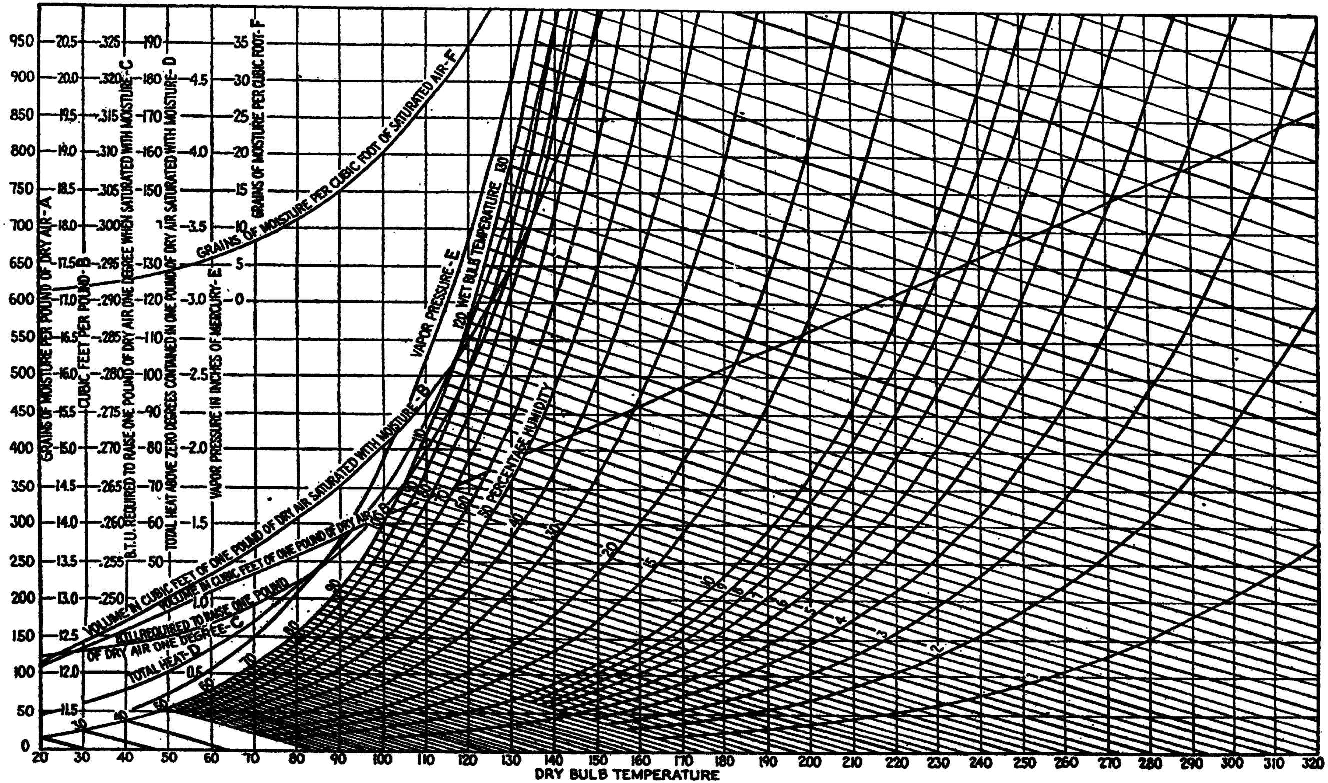

Fig. 1.—Psychrometric chart for high temperatures. Barometric pressure is 29.92 in. of mercury. (*From "Fan Engineering," reproduced by permission of the Buffalo Forge Company.*)

Temp., °F.	C Per cent	Temp., °F.	C Per cent
10	0.21	60	1.76
20	0.34	70	2.50
30	0.55	80	3.51
40	0.83	90	4.86
50	1.22	100	6.67

To illustrate the actual variation that exists, assume air at 70° and 50 per cent relative humidity. According to Eq. (*C*), per cent saturation is then equal to $(0.50) - 0.0250 \times 0.50(1 - 0.50) = 0\,49375$, or 49.375 per cent. The difference 0.625 per cent represents the deviation that exists where Dalton's law is used and the effect of intermolecular forces is neglected.

The refinements previously discussed and the differences cited will hardly be considered sufficiently important to be used in calculations for moist-air conditioning. However, there is a need for such data in order to establish a standard basis for accuracy and thermodynamic consistency. Those desiring a more detailed development of the above subjects may refer to the following recent articles:

"The Interaction Constant for Moist Air," by Goff and Bates, *Trans. A.S.H. & V.E.*, 1941.
"Vapor Pressure of Ice from 32 to −280 F.," by John A. Coff, *Trans. A.S.H. & V.E.*, 1942.
"Final Values of the Interaction Constant for Moist Air," by Goff, Andersen, and Gratch, *Trans. A.S.H. & V.E.*, 1943.
"Thermodynamic Properties of Moist Air," by Goff and Gratch, *Jour. A.S.H. & V.E.*, June, 1945.
"Low Pressure Properties of Water from −160 to 212 F.," by Goff and Gratch, *Jour. A.S.H. & V.E.*, February, 1946.
A.S.H. & V.E. "Guide," 1945, Chap. 1.

TABLE 1.—CAST-IRON RADIATORS, DIMENSIONS AND RATED HEATING SURFACE PER SECTION, SQ. FT.*,†

A.—COLUMN TYPE

Height, inches	One-column	Two-column	Three-column	Four-column	Six-column or "window" pattern
45	. . .	5	6	10	
38	3	4	5	8	
32	2½	3⅓	4½	6½	
26	2	2⅔	3¾	5	
23	1⅔	2⅓			
22	. . .	2¼	3	4	
20	1½	2	. . .	. . .	5
18	. . .	. . .	2¼	3	
16	. . .	. . .	. . .	. . .	3¾
15	. . .	1½			
14					
13	. . .	. . .	. . .	. . . .	3

B.—LARGE TUBE TYPE

Dimensions	Three-tube	Four-tube	Five-tube	Six-tube	Seven-tube
Height, in.‡					
14	. . .	. . . .	. . .	. . . . .	2½
17	. . .	. . . .	. . .	. . . . .	3
20	1¾	2¼	2⅔	3	3⅔
23	2	2½	3	3½	
26	2⅓	2¾	3½	4	
32	3	3½	4⅓	5	
38	3½	4¼	5	6	
Width, in.	4⅝	6 5/16	8	9 11/16	11⅜

NOTE.—Length, 2½ in. per section. Tappings, 1½ in. top and bottom, with bushings as specified. Weight of this type of radiator is about 5.5 lb. per sq. ft. of surface, and internal volume averages 28 cu. in. per sq. ft. of surface.

C.—WALL TYPE§

Height, in.	Length or width, in.	Thickness, in.	Heating surface, sq. ft.
13 5/16	16⅝	2⅞	5
13 5/16	21⅞	2⅞	7
21⅞	13 5/16	3 1/16	7
13 5/16	29 1/16	2⅞	9
29 1/16	13 5/16	3 1/16	9

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* American Radiator Company. Heating-surface data for the modern tube-type radiator are given in Chap. V.

† Based upon a heat emission of 240 B.t.u. per sq. ft. per hr. with steam at a temperature of 215° and the surrounding air at 70°.

‡ Over-all height measured from the floor.

§ Refer to Table 26, Chap. V, for proper heat emission values.

TABLE 2.—CONDUCTIVITIES k AND CONDUCTANCES C OF BUILDING AND INSULATING MATERIALS*

(These constants are expressed in B.t.u. per hr. per sq. ft. per deg. F. temperature difference. Conductivities k are per inch thickness, and conductances C are for thickness or construction stated, not per in. thickness)

Material	Description	Density, lb. per cu. ft.	Mean temp., °F.	Conductivity or conductance		Resistance	Authority
				k	C	$\left(\frac{1}{k}\right)$ or $\left(\frac{1}{C}\right)$	
Building boards (non-insulating).	Compressed cement and asbestos sheets	123	86	2.70	....	0.37	1
	Corrugated asbestos board	20.4	110	0 48	.	2.08	2
	Pressed asbestos millboard	60.5	86	0.84	. ..	1.19	3
	Gypsum board—gypsum between layers of heavy paper	62 8	70	1.41	....	0.71	3
	⅜-in. Gypsum board	......	...	.....	3.73	0.27	
	½-in. Gypsum board	......	...	....	2.82	0 35	
	½-in. Gypsum board	53.5	90	.....	2.60	0 38	1
Frame construction combinations.	1-in. Fir sheathing and building paper	......	30	.....	0.86	1.16	4
	1-in. Fir sheathing, building paper, and yellow-pine lap siding	......	20	.....	0.50	2.00	4
	1-in. Fir sheathing, building paper, and stucco	.....	20	.....	0.82	1.22	4
	Pine lap siding and building paper, siding 4-in. wide	......	16	.....	0.85	1.18	4
	Yellow-pine lap siding	......	...	.....	1.28	0.78	4
Masonry materials:							
Brick	Damp or wet	......	...	5.0‖	....	0.20	2
	Common yellow clay brick*	......	...	4.8	....	0.21	4
	One tier yellow common clay brick, one tier face brick, approx. 8 in. thick*	......	...	.....	0.77	1.30	4
Clay tile, hollow	2-in. Tile, ½-in. plaster both sides	120.0	110	.....	1.00	1.00	2
	4-in. Tile, ½-in. plaster both sides	127.0	100	.....	0.60	1.67	2
	6-in. Tile, ½-in. plaster both sides	124.3	105	.....	0.47	2.13	2
	8-in. Tile, average of 8 types (walls 59, 63, 64, 66, 67, 90, 91, 92)*	......	...	.....	0.52	1.92	4
	12-in. Clay tile wall: 8 in. × 5 in. × 12 in. and 4 in. × 5 in. × 12 in.*	.....	...	.....	0.26	3.84	4
Concrete	Sand and gravel aggregate, various ages and mixes §	......	...	11.35 to 16.36	....	0.09 to 0.06	5
	Sand and gravel aggregate	142	75	12.6	....	0.08	4
	Limestone aggregate	132	75	10.8	....	0.09	4
	Cinder aggregate	97	75	4.9	....	0.22	4
	Steam-treated limestone slag aggregate*	74.6	75	2.27	....	0.44	4

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* See notes on p. 632

TABLE 2.—CONDUCTIVITIES k AND CONDUCTANCES C OF BUILDING AND INSULATING MATERIALS.—(*Continued*)

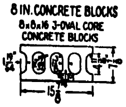
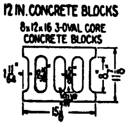
Material	Description	Density, lb. per cu. ft.	Mean temp., °F.	Conductivity or conductance		Resistance	Authority
				k	C	$\left(\frac{1}{k}\right)$ or $\left(\frac{1}{C}\right)$	
Masonry materials —(continued) Concrete— (continued).....	Pumice (mined in California) aggregate*..........................	65.0	75	2.42	....	0.41	4
	Expanded burned-clay aggregate*..	59.9	75	2.28	....	0.44	4
	Burned-clay aggregate*...........	67.1	75	2.86	....	0.35	4
	Blast-furnace slag aggregate.......	76.0	70	1.6	....	0.63	3
	Expanded vermiculite aggregate...	20	90	0.68	....	1.47	3
	Expanded vermiculite aggregate...	26.7	90	0.76	....	1.32	3
	Expanded vermiculite aggregate...	35	90	0.86	....	1.16	3
	Expanded vermiculite aggretate....	50	90	1.10	....	0.91	3
	Concrete plank..................	76	75	2.5	...	0.40	3
	Cellular concrete................	40.0	75	1.06	....	0.94	3
	Cellular concrete................	50.0	75	1.44	....	0.69	3
	Cellular concrete................	60.0	75	1.80	....	0.56	3
	Cellular concrete................	70.0	75	2.18	....	0.46	3
 8 IN. CONCRETE BLOCKS 8x8x16 3-OVAL CORE CONCRETE BLOCKS 15 7/8	8-in. three-oval core, sand and gravel aggregate*...............	126.4	40	.....	0.90	1.11	4
	8-in. three-oval core, crushed limestone aggregate*................	134.3	40	.....	0.86	1.16	4
	8-in. three-oval core, cinder aggregate*..........................	86.2	40	.....	0.58	1.73	4
	8-in. three-oval core, burned-clay aggregate*.....................	67.7	40	.....	0.50	2.00	4
	8-in. three-oval core, expanded blast-furnace slag aggregate*.....	......	40	.....	0.49	2.04	4
 12 IN. CONCRETE BLOCKS 8x12x16 3-OVAL CORE CONCRETE BLOCKS 15 7/8	12-in. three-oval core, sand and gravel aggregate*...............	124.9	40	.....	0.78	1.28	4
	12-in. three-oval core, cinder aggregate*..........................	86.2	40	.....	0.53	1.88	4
	12-in. three-oval core, burned-clay aggregate*....................	76.7	40	.....	0.47	2.13	4
Gypsum.........	3-in. Solid gypsum partition tile*...	......	...	2.41	....	0.42	4
	3-in. Three-cell gypsum partition tile*.........................	......	...	.....	0.74	1.35	4
	4-in. Three-cell gypsum partition tile*.........................	......	...	.....	0.60	1.67	4
	87½ per cent Gypsum, 12½ per cent wood chips................	51.2	74	1.66	....	0.60	4
	Gypsum plaster.................	......	...	3.30	....	0.30	
Plastering materials	Gypsum plaster, ⅜ in. thick......	......	73	.....	8.80	0.11	4
	Cement plaster.................	......	...	8.00	....	0.13	2

TABLE 2.—CONDUCTIVITIES k AND CONDUCTANCES C OF BUILDING AND INSULATING MATERIALS.—(*Continued*)

Material	Description	Density, lb. per cu. ft.	Mean temp., °F.	Conductivity or conductance		Resistance	Authority
				k	C	$\left(\frac{1}{k}\right)$ or $\left(\frac{1}{C}\right)$	
Plastering materials—(continued)....	Wood lath and plaster, total thickness ¾ in..	......	70	.....	2.50	0.40	4
	Gypsum plaster and expanded vermiculite, 4 to 1 mix.............	39.9	75	0.85	....	1.18	3
	Insulating plaster 0.9 in. thick applied to ⅜-in. gypsum board.....	54.0	75	.....	1.07	0.93	3
Roofing............	Asbestos shingles.......	65.0	75	.....	6.0	0.17	3
	Asphalt, composition or prepared..	70.0	75	.....	6.5	0.15	3
	Asphalt shingles.................	70.0	75	.....	6.5	0.15	3
	Built-up roofing, bitumen or felt, gravel or slag surfaced††.........	......	...	1.33	....	0.75	2
	Slate..........................	......	...	10.00	....	0.10	
	Wood shingles..................	......	...	.....	1.28	0.78	
Woods............	Balsa.........................	20.0	90	0.58	....	1.72	1
	Balsa.........................	8.8	90	0.38	....	2.63	1
	Balsa.........................	7.3	90	0.33	....	3.03	1
	California redwood, 0 per cent moisture*.........................	28.0	75	0.70	....	1.43	4
	Cypress.......................	28.7	86	0.67	....	1.49	1
	Douglas fir, 0 per cent moisture*...	34.0	75	0.67	....	1.49	4
	Eastern hemlock, 0 per cent moisture*.........................	30.0	75	0.76	....	1.32	4
	Longleaf yellow pine, 0 per cent moisture*.....................	40.0	75	0.86	....	1.16	4
	Mahogany.....................	34.3	86	0.90	....	1.11	1
	Hard maple, 0 per cent moisture*..	46.0	75	1.05	....	0.95	4
	Maple.........................	44.3	86	1.10	....	0.91	1
	Maple, across grain...............	40.0	75	1.20	....	0.83	3
	Norway pine, 0 per cent moisture*..	32.0	75	0.74	....	1.35	4
	Red cypress, 0 per cent moisture*..	32.0	75	0.79	....	1.27	4
	Red oak, 0 per cent moisture*.....	48.0	75	1.18	....	0.85	4
	Shortleaf yellow pine, 0 per cent moisture*.....................	36.0	75	0.91	....	1.10	4
	Soft elm, 0 per cent moisture*......	34.0	75	0.88	....	1.14	4
	Soft maple, 0 per cent moisture*...	42.0	75	0.95	....	1.05	4
	Sugar pine, 0 per cent moisture*...	28.0	75	0.64	....	1.56	4
	Virginia pine....................	34.3	86	0.96	....	1.04	1
	West coast hemlock, 0 per cent moisture*.....................	30.0	75	0.79	....	1.27	4
	White pine.....................	31.2	86	0.78	....	1.28	1
	Yellow pine....................	.....	...	1.00	....	1.00	3
	Sawdust, various................	12.0	90	0.41	....	2.44	1
	Shavings, various from planer.....	8.8	90	0.41	....	2.44	1
	Shavings, from maple beech and birch (coarse)..................	13.2	90	0.36	....	2.78	1

TABLE 2.—CONDUCTIVITIES k AND CONDUCTANCES C OF BUILDING AND INSULATING MATERIALS.—(*Continued*)

Material	Description	Density, lb. per cu. ft.	Mean temp., °F.	Conductivity or conductance		Resistance	Authority
				k	C	$\left(\frac{1}{k}\right)$ or $\left(\frac{1}{C}\right)$	
Insulating materials Blanket and batt insulations.....	Chemically treated wood fibers held between layers of strong paper....	3.62	70	0.25	....	4.00	3
	Eel grass between strong paper....	4.60	90	0.26	....	8.85	1
	Eel grass between strong paper....	3.40	90	0.25	....	4.00	1
	Flax fibers between strong paper...	4.90	90	0.28	....	3.57	1
	Chemically treated hog hair between kraft paper....................	5.76	71	0.26	....	3.85	3
	Chemically treated hog hair between kraft paper and asbestos paper...	7.70	71	0.28	....	3.57	3
	Hair felt between layers of paper...	11.00	75	0.25	....	4.00	3
	Kapok between burlap or paper....	1.00	90	0.24	....	4.17	1
	Stitched and creped expanding fibrous blanket..................	1.50	70	0.27	....	3.70	3
	Paper and asbestos fiber with emulsified asphalt binder............	4.2	94	0.28	....	3.57	1
	Cotton insulating batt...........	0.875	72	0.24	....	4.17	3
	Felted cattle hair................	13.00	90	0.26	....	3.84	1
	Felted cattle hair................	11.00	90	0.26	....	3.84	1
	Felted hair and asbestos..........	7.80	90	0.28	....	3.57	1
	Ground paper between two layers, each 3/8 in. thick made up of two layers of kraft paper (sample 3/4 in. thick).........................	12.1	75	....	0.40	2.50	4
	3-in. Mineral-wool batts, barrier lapped on warm side; horizontal position†......................	3.67	....	0.30	....	3.33	4
	3-in. Mineral-wool batts, barrier laid on warm side; horizontal position†.	2.24	...	0.26	....	3.84	4
	3-in. Mineral-wool batts, barrier laid on warm side; vertical position†...	2.24	...	0.25	....	4.00	4
	4-in. Mineral-wool batts, barrier lapped on warm side; horizontal position†......................	3.0	...	0.31	....	3.22	4
	4-in. Mineral-wool batts, barrier lapped on warm side; vertical position†........................	3.0	...	0.33	....	3.03	4
	4-in. Mineral-wool batts, no barriers; horizontal position†........	1.77	...	0.30	....	3.33	4
Insulating board....	Made from sugar-cane fiber.......	13.5	70	0.33	....	3.03	3
	Made from cornstalks..	15.00	71	0.33	....	3.03	3
	Made from exploded wood fibers...	17.90	78	0.32	....	3.12	4
	Made from hard wood fibers.......	15.20	70	0.32	....	3.12	3
	Made from wood fiber............	15.90	72	0.33	....	3 03	3
	Made from wood fiber............	15.00	70	0.33	....	3.03	3

TABLE 2.—CONDUCTIVITIES k AND CONDUCTANCES C OF BUILDING AND INSULATING MATERIALS.—(*Continued*)

Material	Description	Density, lb. per cu. ft.	Mean temp., °F.	Conductivity or conductance		Resistance	Authority
				k	C	$\left(\frac{1}{k}\right)$ or $\left(\frac{1}{C}\right)$	
Insulating materials —Continued Insulating board —Continued	Made from wood fiber	......	52	0.33	....	3.03	6
	Made from wood fiber	8.50	72	0.29	....	3.45	3
	Made from wood fiber	15.20	...	0.33	....	3.03	3
	Made from wood fiber	16.30	90	0.34	....	2.94	1
	Made from licorice root	16.1	81	0.34	....	2 94	3
	½-in. insulating boards without special finish** (11 samples)	16.5 to 2[illegible].8	90 ...	0.33 to 0.40	. . ....	3.03 to. 2.40	1
	1-in insulating board*	13.2	...	0.34	...	2.94	4
Loose-fill type	Made from ceiba fibers	1.90	75	0.23	...	4.35	3
	Made from ceiba fibers	1.60	75	0.24	....	4.17	3
	Fibrous material made from dolomite and silica	1.50	75	0.27	....	3.70	3
	Fibrous material made from slag	9.40	103	0.27	....	3.70	1
	Redwood bark	3.00	90	0.31	....	3.22	1
	Redwood bark	5.00	75	0.26	....	3.84	3
	Glass wool fibers 0.0003 to 0.006 in. in diameter	1.50	75	0.27	....	3.70	3
	Granular insulation made from combined silicate of lime and alumina	4.20	72	0.24	....	4.17	3
	Expanded vermiculite	......	...	0.48	....	2.08	1
	Expanded vermiculite, particle size −3 + 14	6.2	..	0.32	....	3.12	3
	Regranulated cork, about $\frac{3}{16}$-in. particles	8.10	90	0.31	....	3.22	1
	Hand-applied granular mineral wool 2 to 6 in. thick; horizontal position† No covering	6.05 to 7.13	... ...	0.30 to 0.33	....	3.33 to 3.03	4
	4-in. machine-blown granular mineral wool, horizontal position.† No covering	5.74	...	0.30	....	3.33	4
	Rock wool	10.0	90	0.27	....	3.70	1
Slab insulations	Corkboard, no added binder	14.0	90	0.34	....	2.94	1
	Corkboard, no added binder	10.6	90	0.30	....	3.33	1
	Corkboard, no added binder	7.0	90	0.27	....	3.70	1
	Corkboard, no added binder	5.4	90	0.25	....	4.00	1
	Corkboard*	8.7	...	0.29	....	3.45	4
	Corkboard, asphaltic binder	14.5	90	0.32	....	3.12	1
	Chemically treated hog hair with film of asphalt	10.0	75	0.28	....	3.57	3
	Sugar-cane fiber insulation blocks encased in asphalt membrane	13.8	70	0.30	....	3.33	3
	Made from 85 per cent magnesia and 15 per cent asbestos	19.3	86	0.51	....	1.96	1
	Made from shredded wood and cement	24.2	72	0.46	....	2.17	3
	Made from shredded wood and cement*	29.8	...	0.77	....	1.30	4

TABLE 2.—CONDUCTIVES k AND CONDUCTANCES C OF BUILDING AND INSULATING MATERIALS.—(*Concluded*)

Authorities:

1. National Bureau of Standards, tests based on samples submitted by manufacturers.
2. A. C. Willard, L. C. Lichty, and L. A. Harding, tests conducted at the University of Illinois.
3. J. C. Peebles, tests conducted at Armour Institute of Technology, based on samples submitted by manufacturers.
4. F. B. Rowley and others, tests conducted at the University of Minnesota.
5. A.S.H. & V.E. Research Laboratory.
6. E. A. Allcut, tests conducted at the University of Toronto.

* F. B. ROWLEY and A. B. ALGREN, "Thermal Conductivity of Building Materials," *Univ. Minn. Eng. Exp. Sta. Bull.* 12.

† F. B. ROWLEY and C. E. LUND, "Heat Transmission through Insulation as Affected by Orientation of Walls," *Trans. A.S.H.& V.E.*, 1943.

‡ G. B. WILKES and L. R. VIANEY, "The Effect of Convection in Ceiling Insulation," *Trans. A.S.H & V.E.*, 1943.

§ F. C. HOUGHTEN and CARL GUTBERLET, "Conductivity of Concrete," *Trans. A.S.H. & V.E.*, 1932.

‖ Recommended value. (See HARDING and WILLARD, "Heating, Ventilating and Air Conditioning," revised ed., 1932.)

** Nat. Bur. Standards, Building Materials and Structures, Report 13.

†† Roofing 0.15 in. thick (1.34 lb. per sq. ft.), covered with gravel (0.83 lb. per sq. ft.), combined thickness assumed 0.25 in.

TABLE 3.—CONDUCTIVITIES k AND CONDUCTANCES C USED IN CALCULATING HEAT-LOSS COEFFICIENTS U IN TABLES 4 TO 15

(These constants are expressed in B.t.u. per hr. per sq. ft. per deg. F. temperature difference. Conductivities k are per in. thickness and conductances C are for thickness or construction stated, not per in. thickness)

Material	Description	Conductivity or conductance		Resistance
		k	C	$\left(\frac{1}{k}\right)$ or $\left(\frac{1}{C}\right)$
Air spaces				
Bounded by ordinary materials	Vertical,* ¾ in. or more in width.	.....	1.10	0.91
Bounded by aluminum foil	Vertical,* ¾ in. or more in width.	.....	0.46	2.17
Exterior finishes (frame walls)				
Brick veneer	4 in. thick (nominal)	.....	....	0.44
Stucco (1 in.)		12.50	....	0.08
Wood shingles		.....	1.28	0.78
Yellow-pine lap siding		.....	1.28	0.78
Insulating materials				
Aluminum foil	See air spaces	.....	....	....
Batts	Enclosed both sides	0 27	....	3.70
Blankets	Made from mineral or vegetable fibers or animal hair	0 27	....	3.70
Corkboard	Pure, no added binder	0.30	....	3.33
Insulating board		0.33	....	3.03
Mineral wool		0.27	....	3.70
Vermiculite		0.48	....	2.08
Interior finishes				
Composition wallboard	3/16 to ⅜ in. thick	0.50	....	2.00
Gypsum plaster		3.30	....	0.30
Gypsum board (⅜ in.)	Plain or decorated	.....	3.70	0.27
Gypsum lath (⅜ in.) and plaster	Plaster thickness assumed ½ in.	.....	2.4	0.42
Insulating board (½ in.)	Plain or decorated	.....	....	1.52
Insulating-board lath (½ in.) and plaster	Plaster thickness assumed ½ in.	.....	....	1.67
Insulating-board lath (1 in.) and plaster	Plaster thickness assumed ½ in.	.....	....	3.18
Metal lath and plaster	Plaster thickness assumed ¾ in.	.....	4.40	0.23
Plywood (⅜ in.)	Plain or decorated	.....	....	0.47
Wood lath and plaster		.....	2.50	0.40
Masonry materials				
Brick	Adobe	3.56	....	0.28
Brick	Common	5.00	....	0.20
Brick	Face	9.20	....	0.11
Cement mortar		12.00	....	0.08
3-in. clay tile (hollow)		.....	1.28	0.78
4-in. clay tile (hollow)		.....	1.00	1.00
6-in. clay tile (hollow)		.....	0.64	1.57

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* Conductance values for horizontal air spaces depend on whether the heat flow is upward or downward, but in most cases it is sufficiently accurate to use the same values for horizontal as for vertical air spaces.

† Expanded slag, burned clay, or pumice.

TABLE 3.—CONDUCTIVITIES k AND CONDUCTANCES C USED IN CALCULATING HEAT-LOSS COEFFICIENTS U IN TABLES 4 TO 15.—(*Concluded*)

Material	Description	Conductivity or conductance		Resistance
		k	C	$\left(\frac{1}{k}\right)$ or $\left(\frac{1}{C}\right)$
Masonry Materials				
8-in. clay tile (hollow)		.....	0.60	1.67
10-in. clay tile (hollow)		.....	0.58	1.72
12-in. clay tile (hollow)		.....	0.40	2.50
16-in. clay tile (hollow)		....	0.31	3.23
Concrete	Lightweight aggregate†	2.50	....	0.40
Concrete	Sand and gravel aggregate	12.00	....	0.08
3-in. concrete blocks	Hollow, cinder aggregate	....	1.28	0.78
4-in. concrete blocks	Hollow, cinder aggregate	.....	1.00	1.00
8-in. concrete blocks	Hollow, gravel aggregate	.....	1.00	1.00
12-in. concrete blocks	Hollow, gravel aggregate	.....	0.80	1.25
8-in. concrete blocks	Hollow, cinder aggregate	.....	0.60	1.66
12-in. concrete blocks	Hollow, cinder aggregate	.....	0.53	1.88
8-in. concrete blocks	Hollow, lightweight aggregate†	.....	0.50	2.00
12-in. concrete blocks	Hollow, lightweight aggregate†	.....	0.47	2.13
Gypsum fiber concrete	87½ per cent gypsum and 12½ per cent wood chips	1.66	....	0.60
3 in. gypsum tile	Hollow	.....	0.61	1.64
4-in. gypsum tile	Hollow	.....	0.46	2.18
Stucco		12.50	.....	0.08
Tile and terrazzo	For flooring	12.00	....	0.08
Roofing materials				
Asbestos shingles		....	6.00	0.17
Asphalt shingles		.....	6.50	0.15
Built-up roofing	Assumed thickness ⅜ in.	.....	3.53	0.28
Heavy roll roofing		.....	6.50	0.15
Slate		10.00	.....	0.10
Wood shingles		.....	1.28	0.78
Sheathing				
Gypsum (½ in.)		.....	2.82	0.35
Insulating board (²⁵⁄₃₂ in.)		.....	....	2.37
Plywood (⁵⁄₁₆ in.)		.....	....	0.39
Fir or yellow pine (1 in.)	Actual thickness ²⁵⁄₃₂ in.	.....	....	0.98
Fir, plus building paper	Actual thickness ²⁵⁄₃₂ in.	.....	0.86	1.16
Surfaces				
Still air	Ordinary nonreflective materials, vertical	.....	1.65	0.61
15 m.p.h. wind velocity	Ordinary nonreflective materials, vertical	.....	6.00	0.17
Woods				
Fir sheathing (1 in.) building paper and yellow-pine lap siding		.....	0.50	2.00
Maple or oak		1.15	....	0.87
Yellow pine or fir		0.80	...	1.25

The thicknesses upon which the coefficients in Tables 4 to 16 inclusive are based are as follows:

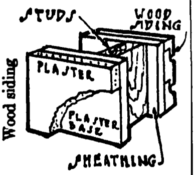
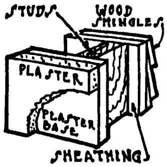
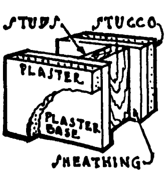
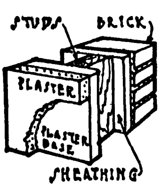
Brick veneer	4 in.
Plaster and metal lath	3/4 in.
Plaster (on wood lath, gypsum lath, or insulating board)	1/2 in.
Slate (roofing)	1/2 in.
Stucco on wire-mesh reinforcing	1 in.
Tar and gravel or slag-surfaced built-up roofing	3/8 in.
1-in. lumber (S-2-S)	25/32 in.
1 1/2-in. lumber (S-2-S)	1 5/16 in.
2-in. lumber (S-2-S)	1 5/8 in.
2 1/2-in. lumber (S-2-S)	2 1/8 in.
3-in. lumber (S-2-S)	2 5/8 in.
4-in. lumber (S-2-S)	3 5/8 in.
Finish flooring (maple or oak)	13/16 in.

The coefficients of transmission computed in these tables for solid-brick walls are based on 4 in. hard brick (high density) and the remainder common brick (low density). When stucco is applied to masonry walls it is assumed to be 1 in. thick. The metal lath is neglected where metal lath and plaster are specified.

Calculated coefficients involving lumber use actual thicknesses instead of nominal thicknesses. For wood shingle roofs applied over wood stripping, the computations are based on 1 by 4 in. wood strips, spaced 2 in. apart.

The pitched-roof coefficients in Table 13 apply where the roof is over a heated attic or top floor with the heat flowing directly through the roof, including whatever finish is applied to the underside of the roof rafters. Values for Spanish and French clay roofing tile are assumed to be the same as for slate roofs.

TABLE 4.—COEFFICIENTS OF TRANSMISSION U OF FRAME WALLS
(Coefficients are expressed in B.t.u. per hr. per sq. ft. per deg. F. difference in temperature between the air on the two sides, and are based on a wind velocity of 15 m.p.h. No insulation between studs* (see Table 5)

Exterior finish	Interior finish	Type of sheathing				Wall number
		Gyp-sum (½ in. thick)	Ply-wood (5/16 in. thick)	Wood† (25/32 in. thick) bldg. paper	Insu-lating board (25/32 in. thick)	
		A	B	C	D	
Wood siding  STUDS WOOD SIDING PLASTER PLASTER BASE SHEATHING	Metal lath and plaster‡	0.33	0.32	0.26	0.20	1
	Gypsum board (⅜ in.) decorated	0.32	0.32	0.25	0.20	2
	Wood lath and plaster	0.31	0.31	0.25	0.19	3
	Gypsum lath (⅜ in.), plastered§	0.31	0.31	0.25	0.19	4
	Plywood (⅜ in.), plain or decorated	0.30	0.30	0.24	0.19	5
	Insulating board (½ in.), plain or decorated	0.23	0.23	0.19	0.16	6
	Insulating-board lath (½ in.), plastered§	0.22	0.22	0.19	0.15	7
	Insulating-board lath (1 in.), plastered§	0.17	0.17	0.15	0.12	8
Wood[d] shingles  STUDS WOOD SHINGLES PLASTER PLASTER BASE SHEATHING	Metal lath and plaster‡	0.25	0.25	0.26	0.17	9
	Gypsum board (⅜ in.), decorated	0.25	0.25	0.25	0.17	10
	Wood lath and plaster	0.24	0.24	0.25	0.16	11
	Gypsum lath (⅜ in.), plastered§	0.24	0.24	0.25	0.16	12
	Plywood (⅜ in.), plain or decorated	0.24	0.24	0.24	0.16	13
	Insulating board (½ in.), plain or decorated	0.19	0.19	0.19	0.14	14
	Insulating-board lath (½ in.), plastered§	0.19	0.18	0.19	0.13	15
	Insulating-board lath (1 in.), plastered§	0.14	0.14	0.15	0.11	16
Stucco 	Metal lath and plaster‡	0.43	0.42	0.32	0.23	17
	Gypsum board (⅜ in.), decorated	0.42	0.41	0.31	0.23	18
	Wood lath and plaster	0.40	0.39	0.30	0.22	19
	Gypsum lath (⅜ in.), plastered§	0.39	0.39	0.30	0.22	20
	Plywood (⅜ in.), plain or decorated	0.39	0.38	0.29	0.22	21
	Insulating board (½ in.), plain or decorated	0.27	0.27	0.22	0.18	22
	Insulating-board lath (½ in.), plastered§	0.26	0.26	0.22	0.17	23
	Insulating-board lath (1 in.), plastered§	0.19	0.19	0.16	0.14	24
Brick veneer[e] 	Metal lath and plaster‡	0.37	0.36	0.28	0.21	25
	Gypsum board (⅜ in.), decorated	0.36	0.36	0.28	0.21	26
	Wood lath and plaster	0.35	0.34	0.27	0.20	27
	Gypsum lath (⅜ in.), plastered§	0.34	0.34	0.27	0.20	28
	Plywood (⅜ in.), plain or decorated	0.34	0.33	0.27	0.20	29
	Insulating board (½ in.), plain or decorated	0.25	0.25	0.21	0.17	30
	Insulating-board lath (½ in.), plastered§	0.24	0.24	0.20	0.16	31
	Insulating-board lath (1 in.), plastered§	0.18	0.18	0.15	0.13	32

* Coefficients not weighted; effect of studding neglected.

† Nominal thickness, 1 in.

‡ Plaster assumed ¾ in. thick.

§ Plaster assumed ½ in. thick.

[d] Furring strips between wood shingles and all sheathings except wood.

[e] Small air space and mortar between building paper and brick veneer neglected.

TABLE 5.—COEFFICIENTS OF TRANSMISSION *U* OF FRAME WALLS WITH INSULATION BETWEEN FRAMING*,†

Coefficients are expressed in B.t.u. per sq. ft. per deg. F. difference in temperature between the air on the two sides, and are based on a wind velocity of 15 m.p.h.

Coefficient with no insulation between framing	Coefficient with insulation between framing				Wall number
	Blanket or batt insulation between framing‡ (thickness below)			$3\frac{5}{8}$-in. loose mineral wool between framing	
	1 in.	2 in.	3 in.		
	A	B	C	D.	
0.11	0.078	0.064	0.055	0.051	33
0.12	0.083	0.067	0.057	0.054	34
0.13	0.088	0.070	0.059	0.056	35
0.14	0.092	0.073	0.061	0.058	36
0.15	0.097	0.075	0.062	0.059	37
0.16	0.10	0.077	0.065	0.060	38
0.17	0.10	0.080	0.066	0.062	39
0.18	0.11	0.082	0.068	0.063	40
0.19	0.11	0.084	0.069	0.064	41
0.20	0.12	0.087	0.070	0.066	42
0.21	0.12	0.088	0.072	0.067	43
0.22	0.12	0.090	0.073	0.069	44
0.23	0.12	0.093	0.074	0.069	45
0.24	0.12	0.094	0.076	0.070	46
0.25	0.13	0.095	0.076	0.072	47
0.26	0.13	0.096	0.077	0.072	48
0.27	0.14	0.097	0.078	0.073	49
0.28	0.14	0.098	0.078	0.073	50
0.29	0.14	0.10	0.080	0.075	51
0.30	0.14	0.10	0.080	0.075	52
0.31	0.14	0.10	0.082	0.076	53
0.32	0.15	0.10	0.082	0.076	54
0.33	0.15	0.11	0.083	0.077	55
0.34	0.15	0.11	0.083	0.078	56
0.35	0.15	0.11	0.085	0.078	57
0.36	0.16	0.11	0.085	0.079	58
0.37	0.16	0.11	0.087	0.080	59
0.38	0.16	0.11	0.087	0.080	60
0.39	0.16	0.11	0.087	0.081	61
0.40	0.16	0.11	0.088	0.082	62
0.41	0.16	0.11	0.088	0.082	63
0.42	0.16	0.11	0.088	0.082	64
0.43	0.17	0.11	0.090	0.083	65
0.44	0.17	0.12	0.090	0.083	66

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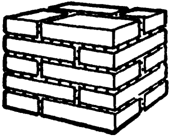
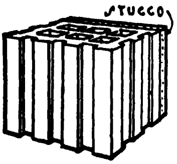
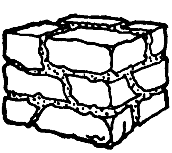
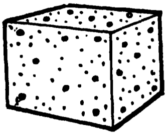
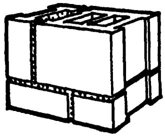
* This table may be used for determining the coefficients of transmission of frame constructions with the types and thicknesses of insulation indicated in columns A to D inclusive between framing. Columns A, B, and C may be used for walls, ceilings, or roofs with only one air space between framing but are not applicable to ceilings with *no* flooring above. (See Table 10.) Column D is applicable to walls only. *Example:* Find the coefficient of transmission of a frame wall consisting of wood siding, $2\frac{5}{32}$-in. insulating-board sheathing, studs, gypsum lath, and plaster, with 2-in. blanket insulation between studs. According to Table 4, a wall of this construction with *no insulation* between studs has a coefficient of 0.19 (wall no. 4D). Referring to column B above, it will be found that a wall of this value with 2-in. blanket insulation between the studs has a coefficient of 0.084.

† Coefficients corrected for 2 × 4 framing, 16 in. on center.

‡ Based on one air space between framing.

TABLE 6.—COEFFICIENTS OF TRANSMISSION U OF MASONRY WALLS (Coefficients are expressed in B.t.u. per hr. per sq. ft. per deg. F. difference in temperature between the air on the two sides, and are based on a wind velocity of 15 m.p.h.)

Type of masonry		Thickness of masonry, in.	Interior finish (plus insulation where indicated) Plain walls—no interior finish	Plaster (½ in.) on walls	Metal lath and plaster* furred	Gypsum board (⅜ in.), decorated and furred	Gypsum lath (⅜ in.), plastered† and furred	Insulating board (½ in.), plain or decorated, furred	Insulating-board lath (½ in.), plastered† and furred	Insulating-board lath (1 in.), plastered† and furred	Gypsum lath†, plastered, plus 1 in. blanket insulation, furred‡	Wall number
			A	B	C	D	E	F	G	H	I	
Solid[a] brick		8	0.50	0.46	0.32	0.31	0.30	0.22	0.22	0.16	0.14	67
		12	0.35	0.34	0.25	0.25	0.24	0.19	0.19	0.14	0.13	68
		16	0.28	0.27	0.21	0.21	0.20	0.17	0.16	0.13	0.12	69
Hollow[b] tile (Stucco Exterior Finish)		8	0.40	0.37	0.27	0.27	0.26	0.20	0.20	0.15	0.13	70
		10	0.39	0.37	0.27	0.27	0.26	0.20	0.19	0.15	0.13	71
		12	0.30	0.29	0.22	0.22	0.21	0.17	0.17	0.13	0.12	72
		16	0.25	0.24	0.19	0.19	0.19	0.16	0.15	0.12	0.11	73
Stone[c]		8	0.70	0.64	0.39	0.38	0.36	0.26	0.25	0.18	0.16	74
		12	0.57	0.53	0.35	0.34	0.33	0.24	0.23	0.17	0.15	75
		16	0.49	0.45	0.32	0.31	0.30	0.22	0.22	0.16	0.14	76
		24	0.37	0.35	0.26	0.26	0.25	0.20	0.19	0.15	0.13	77
Poured concrete[d]		6	0.79	0.71	0.42	0.41	0.39	0.27	0.26	0.19	0.16	78
		8	0.70	0.64	0.39	0.38	0.36	0.26	0.25	0.18	0.16	79
		10	0.63	0.58	0.37	0.36	0.34	0.25	0.24	0.18	0.15	80
		12	0.58	0.53	0.35	0.34	0.33	0.24	0.23	0.17	0.15	81
Hollow concrete blocks		Gravel Aggregate										
		8	0.56	0.52	0.34	0.34	0.32	0.24	0.19	0.17	0.15	82
		12	0.50	0.46	0.32	0.31	0.30	0.22	0.22	0.16	0.14	83
		Cinder Aggregate										
		8	0.41	0.39	0.28	0.28	0.27	0.21	0.20	0.15	0.13	84
		12	0.38	0.36	0.26	0.26	0.25	0.20	0.19	0.15	0.13	85
		Lightweight Aggregate§										
		8	0.36	0.34	0.26	0.25	0.24	0.19	0.19	0.15	0.13	86
		12	0.34	0.33	0.25	0.24	0.24	0.19	0.18	0.14	0.13	87

[a] Based on 4 in. hard brick and remainder common brick. ·

[b] The 8-in. and 10-in. tile figures are based on two cells in the direction of heat flow. The 12-in. tile is based on three cells in the direction of heat flow. The 16-in. tile consists of one 10-in. and one 6-in. tile each having two cells in the direction of heat flow.

[c] Limestone or sandstone.

[d] These figures may be used with sufficient accuracy for concrete walls with stucco exterior finish.

§ Expanded slag, burned clay or pumice.

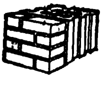
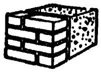
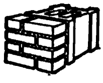
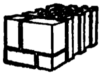
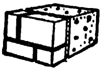
* Thickness of plaster assumed ¾ in.

† Thickness of plaster assumed ½ in.

‡ Based on 2-in. furring strips; one air space. These figures may also be used for wood or metal lath.

TABLE 7.—COEFFICIENTS OF TRANSMISSION U OF BRICK AND STONE VENEER MASONRY WALLS

(Coefficients are expressed in B.t.u. per hr. per sq. ft. per deg. F. difference in temperature between the air on the two sides, and are based on a wind velocity of 15 m.p.h.)

Typical construction	Facing	Backing	Interior finish (plus insulation where indicated)									Wall number
			Plain walls—no interior finish A	Plaster (½ in.) on walls B	Metal lath and plaster,* furred C	Gypsum board (⅜ in.), decorated and furred D	Gypsum lath (⅜ in.) plastered† and furred E	Insulating board (½ in.), plain or decorated, furred F	Insulating-board lath (½ in.), plastered†—furred G	Insulating board lath (1 in.) plastered†—furred H	Gypsum lath plastered† plus 1 in. blanket insulation—furred‡ I	
	4-in. Brick veneer‖	6-in. Hollow tile§	0.35	0.34	0.25	0.25	0.24	0.19	0.18	0.14	0.13	88
		8-in. Hollow tile§	0.34	0.32	0.25	0.24	0.23	0.19	0.18	0.14	0.13	89
		6-in. Concrete	0.59	0.54	0.35	0.35	0.33	0.24	0.23	0.17	0.15	90
		8-in. Concrete	0.54	0.50	0.33	0.33	0.31	0.23	0.23	0.17	0.15	91
		8-in. Concrete blocks** (gravel aggregate)	0.44	0.41	0.29	0.29	0.28	0.21	0.21	0.16	0.14	92
		8-in. Concrete blocks** (cinder aggregate)	0.34	0.33	0.25	0.24	0.24	0.19	0.18	0.14	0.13	93
		8-in. Concrete blocks** (lightweight aggregate)††	0.31	0.29	0.23	0.23	0.22	0.18	0.17	0.14	0.12	94
	4-in. Cut stone veneer‖	6-in. Hollow tile§	0.37	0.35	0.26	0.26	0.25	0.19	0.19	0.15	0.13	95
		8-in. Hollow tile§	0.36	0.34	0.25	0.25	0.24	0.19	0.19	0.15	0.13	96
		6-in. Concrete	0.63	0.58	0.37	0.36	0.34	0.25	0.24	0.18	0.15	97
		8-in. Concrete	0.57	0.53	0.35	0.34	0.33	0.24	0.23	0.17	0.15	98
		8-in. Concrete blocks** (gravel aggregate)	0.47	0.44	0.30	0.30	0.29	0.22	0.21	0.16	0.14	99
		8-in. Concrete blocks** (cinder aggregate)	0.36	0.34	0.25	0.25	0.24	0.19	0.19	0.15	0.13	100
		8-in. Concrete blocks** (lightweight aggregate)††	0.32	0.30	0.23	0.23	0.22	0.18	0.18	0.14	0.12	101

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* Thickness of plaster assumed ¾ in.

† Thickness of plaster assumed ½ in.

‡ Based on 2 in. furring strips; one air space. The figures in this column may also be used for wood or metal lath.

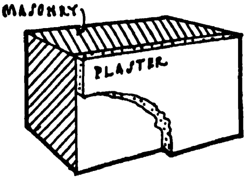
§ The hollow-tile figures are based on two air cells in the direction of heat flow.

‖ Calculations based on ½ in. cement mortar between backing and facing except in the case of the concrete backing, which is assumed to be poured in place.

** Hollow concrete blocks.

†† Expanded slag, burned clay or pumice.

TABLE 8.—COEFFICIENTS OF TRANSMISSION U OF MASONRY PARTITIONS
(Coefficients are expressed in B.t.u. per hr. per sq. ft. per deg. F. difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides)

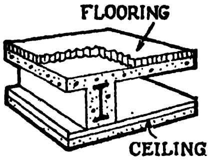
Type of partition 		Thickness of masonry, in.	Type of finish No finish (plain walls) A	 Plaster one side B	 Plaster both sides* C	Partition number
Hollow clay tile		3	0.50	0.47	0.43	9
		4	0.45	0.42	0.40	10
Hollow gypsum tile		3	0.35	0.33	0.32	11
		4	0.29	0.28	0.27	12
Hollow concrete tile or blocks	Cinder aggregate	3	0.50	0.47	0.43	13
		4	0.45	0.42	0.40	14
	Lightweight aggregate†	3	0.41	0.39	0.37	15
		4	0.36	0.34	0.32	16
Common brick		4	0.50	0.46	0.43	17

* 2 in. solid plaster partition, $U = 0.53$.

† Expanded slag, burned clay or pumice.

TABLE 9.—COEFFICIENTS OF TRANSMISSION U OF CONCRETE-CONSTRUCTION FLOORS AND CEILINGS

(Coefficients are expressed in B.t.u. per hr. per sq. ft. per deg. F. difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides)

	Thickness of concrete, (in.)§	Type of flooring					Number
		No flooring (concrete bare)	Tile* or terrazzo flooring on concrete	¼-in. Battleship† linoleum directly on concrete	Parquet‡ flooring in mastic on concrete	Double wood floor on sleepers‖	
		A	B	C	D	E	
No ceiling	3	0.69	0.65	0.45	0.45	0.25	1
	6	0.59	0.56	0.41	0.41	0.23	2
	10	0.50	0.48	0.36	0.36	0.22	3
½-in. Plaster applied to under side of concrete	3	0.62	0.59	0.43	0.43	0.24	4
	6	0.54	0.52	0.39	0.39	0.22	5
	10	0.46	0.44	0.34	0.34	0.21	6
Metal lath and plaster,** suspended or furred	3	0.38	0.37	0.30	0.30	0.19	7
	6	0.35	0.34	0.28	0.28	0.18	8
	10	0.32	0.31	0.26	0.26	0.17	9
Gypsum board (⅜ in.) and plaster,†† suspended or furred	3	0.36	0.35	0.28	0.28	0.19	10
	6	0.33	0.32	0.27	0.27	0.18	11
	10	0.30	0.29	0.24	0.24	0.17	12
Insulating-board lath (½ in.) and plaster,†† suspended or furred	3	0.25	0.24	0.21	0.21	0.15	13
	6	0.23	0.23	0.20	0.20	0.15	14
	10	0.22	0.21	0.19	0.19	0.14	15

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* Thickness of tile assumed to be 1 in.

† The figures in column C may be used with sufficient accuracy for concrete floors covered with carpet.

‡ Thickness of wood assumed to be 13/16 in.; thickness of mastic, ⅛ in. ($k = 4.5$).

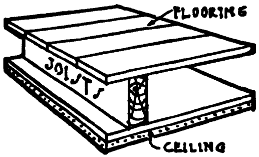
‖ Based on 25/32-in. yellow pine or fir subflooring and 13/16-in. hardwood-finish flooring with an air space between subfloor and concrete.

** Thickness of plaster assumed to be ¾ in.

†† Thickness of plaster assumed to be ½ in.

§ Interpolate for other thicknesses or concrete.

TABLE 10.—COEFFICIENTS OF TRANSMISSION U OF FRAME-CONSTRUCTION CEILINGS AND FLOORS
(Coefficients are expressed in B.t.u. per hr. per sq. ft. per deg. F. difference in temperature between the air on the two sides and are based on still air (no wind) conditions on both sides)

Type of ceiling 	Insulation between or on top of joists (no flooring above)												With flooring* (on top of ceiling joists)		Number
	None	Insulating board on top of joists		Blanket or batt insulation* between joists‡			Vermiculite insulation between joists			Mineral-wool insulation between joists			Single wood floor§	Double wood floor‖	
		½ in.	1 in.	1 in.	2 in.	3 in.	2 in.	3 in.	4 in.	2 in.	3 in.	4 in.			
	A	B	C	D	E	F	G	H	I	J	K	L	M	N	
No ceiling		0.37	0.24										0.45	0.34	1
Metal lath and plaster**	0.69	0.26	0.19	0.19	0.12	0.092	0.18	0.14	0.11	0.12	0.092	0.066	0.30	0.25	2
Gypsum board (⅜ in.), plain or decorated	0.67	0.26	0.18	0.19	0.12	0.092	0.18	0.14	0.10	0.12	0.092	0.066	0.30	0.24	3
Wood lath and plaster	0.62	0.25	0.18	0.19	0.12	0.092	0.17	0.14	0.10	0.12	0.092	0.065	0.29	0.24	4
Gypsum lath (⅜ in.), plastered††	0.61	0.25	0.18	0.19	0.12	0.092	0.17	0.14	0.10	0.12	0.092	0.065	0.28	0.24	5
Plywood (⅜ in.), plain or decorated	0.59	0.24	0.18	0.19	0.12	0.09	0.17	0.14	0.10	0.12	0.090	0.065	0.28	0.23	6
Insulating board (½ in.), plain or decorated	0.37	0.19	0.15	0.16	0.11	0.082	0.15	0.11	0.092	0.11	0.082	0.060	0.22‡‡	0.19‡‡	7
Insulating-board lath (½ in.), plastered††	0.35	0.19	0.15	0.15	0.10	0.081	0.14	0.11	0.091	0.10	0.082	0.060	0.21	0.18	8
Insulating-board (1 in.), plastered††	0.23	0.15	0.12	0.12	0.089	0.072	0.12	0.097	0.080	0.089	0.072	0.055	0.16	0.14	9

* For coefficients for construction in columns M and N (except No. 1) with insulation between joists, refer to Table 5. *Example:* The coefficient for No. 3-N of Table 10 is 0.24. With 2-in. blanket insulation between joists, the coefficient will be 0.094. (See Table 5.) (Column D of Table 5 applicable only for 3⅝-in. joists.)

† Based on insulation in contact with ceiling and consequently no air space between.

‡ Coefficients corrected for framing.

§ $\frac{25}{32}$ in. yellow pine or fir.

‖ $\frac{25}{32}$ in. pine or fir subflooring plus $\frac{13}{16}$ in. hardwood-finish flooring.

** Plaster assumed ¾ in. thick.

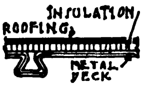
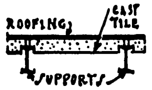
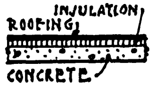
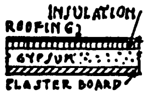
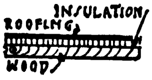
†† Plaster assumed ½ in. thick.

‡‡ For $\frac{25}{32}$-in. insulating-board sheathing applied to the under side of the joists, the coefficient for *single* wood floor (column M) is 0.18 and for *double* wood floor (column N) is 0.16. For coefficients with insulation between joists, see Table 5.

TABLE 11.—COEFFICIENTS OF TRANSMISSION U OF FLAT ROOFS COVERED WITH BUILT-UP ROOFING. NO CEILING—UNDERSIDE OF ROOF EXPOSED

(See Table 12 for flat roofs with ceilings)

(These coefficients are expressed in B.t.u. per hr. per sq. ft. per deg. F. difference in temperature between the air on the two sides, and are based on an outside wind velocity of 15 m.p.h.)

Type of roof deck	Thickness of roof deck, in.	No insulation	Insulation on top of deck (covered with built-up roofing)							Number
			Insulating board (thickness below)				Corkboard (thickness below)			
			½ in.	1 in.	1½ in.	2 in.	1 in.	1½ in.	2 in.	
		A	B	C	D	E	F	G	H	
Flat metal roof deck* 		0.94	0.39	0.24	0.18	0.14	0.23	0.17	0.13	1
Precast cement tile 	1⅝ in.	0.84	0.37	0.24	0.18	0.14	0.22	0.16	0.13	2
Concrete 	2 in.	0.82	0.37	0.24	0.17	0.14	0.22	0.16	0.13	3
	4 in.	0.72	0.34	0.23	0.17	0.13	0.21	0.16	0.12	4
	6 in.	0.65	0.33	0.22	0.16	0.13	0.21	0.15	0.12	5
Gypsum and wood fiber† on ⅜ in. gypsum board 	2⅜ in.	0.40	0.25	0.18	0.14	0.12	0.17	0.13	0.11	6
	3⅜ in.	0.32	0.22	0.16	0.13	0.11	0.15	0.12	0.10	7
Wood‡ 	1 in.	0.49	0.28	0.20	0.15	0.12	0.19	0.14	0.12	8
	1½ in.	0.37	0.24	0.18	0.14	0.11	0.17	0.13	0.11	9
	2 in.	0.32	0.22	0.16	0.13	0.11	0.16	0.12	0.10	10
	3 in.	0.23	0.17	0.14	0.11	0.096	0.13	0.11	0.091	11

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* Coefficient of transmission of bare corrugated iron (no roofing) is 1.50 B.t.u. per hr. per sq. ft. of projected area per deg. F. difference in temperature, based on an outside wind velocity of 15 m.p.h.

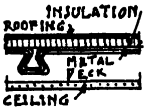
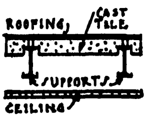
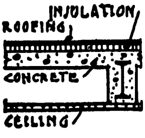
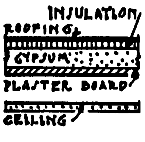
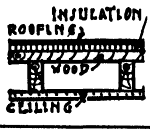
† 87½ per cent gypsum, 12½ per cent wood fiber. Thickness indicated includes ⅜ in. gypsum board.

‡ Nominal thicknesses specified—actual thicknesses used in calculations.

TABLE 12.—COEFFICIENTS OF TRANSMISSION U OF FLAT ROOFS COVERED WITH BUILT-UP ROOFING. WITH LATH AND PLASTER CEILINGS*

(See Table 11 for flat roofs with no ceilings)

(These coefficients are expressed in B.t.u. per hr. per sq. ft. per deg. F. difference in temperature between the air on the two sides, and are based on an outside wind velocity of 15 m.p.h.)

Type of roof deck		Thickness of roof deck, in.	No insulation	Insulation on top of deck (covered with built-up roofing)							Number
				Insulating board (thickness below)				Corkboard (thickness below)			
				½ in.	1 in.	1½ in.	2 in.	1 in.	1½ in.	2 in.	
			A	B	C	D	E	F	G	H	
Flat metal roof deck			0.46	0.27	0.19	0.15	0.12	0.18	0.14	0.11	12
Precast cement tile		1⅝ in.	0.43	0.26	0.19	0.15	0.12	0.18	0.14	0.11	13
Concrete		2 in.	0.42	0.26	0.19	0.15	0.12	0.18	0.14	0.11	14
		4 in.	0.40	0.25	0.18	0.14	0.12	0.17	0.13	0.11	15
		6 in.	0.37	0.24	0.18	0.14	0.11	0.17	0.13	0.11	16
Gypsum and wood fiber† on ⅜ in. gypsum board		2⅜ in.	0.27	0.19	0.15	0.12	0.10	0.14	0.12	0.097	17
		3⅜ in.	0.23	0.17	0.14	0.11	0.097	0.13	0.11	0.091	18
Wood*		1 in.	0.32	0.21	0.16	0.13	0.11	0.15	0.12	0.10	19
		1½ in.	0.26	0.19	0.15	0.12	0.10	0.14	0.11	0.095	20
		2 in.	0.24	0.17	0.14	0.11	0.097	0.13	0.11	0.092	21
		3 in.	0.18	0.14	0.12	0.10	0.087	0.11	0.096	0.082	22

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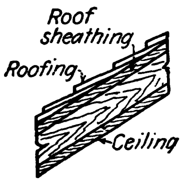
* Calculations based on metal lath and plaster ceilings, but coefficients may be used with sufficient accuracy for gypsum lath or wood lath and plaster ceilings. It is assumed that there is an air space between the under side of the roof deck and the upper side of the ceiling.

† 87½ per cent gypsum, 12½ per cent wood fiber. Thickness indicated includes ⅜ in. gypsum board.

‡ Nominal thicknesses specified—actual thicknesses used in calculations.

TABLE 13.—COEFFICIENTS OF TRANSMISSION U OF PITCHED ROOFS

(Coefficients are expressed in B.t.u. per hr. per sq. ft. per deg. F. difference in temperature between the air on the two sides and are based on a wind velocity of 15 m.p.h.)



	Wood shingles (On 1 × 4 wood strips† spaced 2 in. apart)				Asphalt shingles or Roll roofing (on solid wood sheathing)†				Slate or tile* (on solid wood sheathing)†				Number
	Insulation between rafters				Insulation between rafters				Insulation between rafters				
	None	Blanket or batt (thickness below)			None	Blanket or batt (thickness below)			None	Blanket or batt (thickness below)			
		1 in.	2 in.	3 in.		1 in	2 in.	3 in.		1 in.	2 in.	3 in.	
	A	B	C‡	D‡	E	F	G‡	H‡	I	J	K‡	L‡	
No ceiling applied to rafters	0.48§	0.15	0.11	0.083	0.53§	0.15	0.11	0.085	0.55§	0.16	0.11	0.085	1
Metal lath and plaster‖	0.31	0.14	0.10	0.082	0.33	0.15	0.10	0.083	0.34	0.15	0.11	0.083	2
Gypsum board (⅜ in.) decorated	0.31	0.14	0.10	0.082	0.32	0.15	0.10	0.082	0.34	0.15	0.11	0.083	3
Wood lath and plaster	0.30	0.14	0.10	0.080	0.31	0.14	0.10	0.082	0.32	0.15	0.10	0.082	4
Gypsum lath (⅜ in.), plastered**	0.29	0.14	0.10	0.080	0.31	0.14	0.10	0.082	0.32	0.15	0.10	0.082	5
Plywood (⅜ in.), plain or decorated	0.29	0.14	0.10	0.080	0.30	0.14	0.10	0.082	0.31	0.14	0.10	0.082	6
Insulating board (½ in.), plain or decorated	0.22	0.12	0.090	0.073	0.23	0.12	0.093	0.074	0.24	0.13	0.094	0.076	7
Insulating-board lath (½ in.), plastered**	0.22	0.12	0.090	0.073	0.22	0.12	0.090	0.073	0.23	0.12	0.093	0.074	8
Insulating-board lath (1 in.), plastered**	0.16	0.10	0.077	0.065	0.17	0.10	0.077	0.065	0.17	0.10	0.080	0.066	9

‡ Coefficients corrected for framing.

* Figures in Columns I, J, K, and L may be used with sufficient accuracy for rigid asbestos shingles on wood sheathing. Layer of slater's felt neglected.

† Sheathing and wood strips assumed $^{25}/_{32}$ in. thick.

‖ Plaster assumed ¾ in. thick.

** Plaster assumed ½ in. thick.

§ No air space included in 1-A, 1 – E, or 1-I; all other coefficients based on one air space.

TABLE 14.—COMBINED COEFFICIENTS OF TRANSMISSION *U* OF PITCHED ROOFS* AND HORIZONTAL CEILINGS—BASED ON CEILING AREA†

(Coefficients are expressed in B.t.u. per hr. per sq. ft. of ceiling area per deg. F. difference in temperature between the air on the two sides, and are based on a wind velocity of 15 m.p.h.)

Ceiling coefficient** (from Table 10)	Type of roofing and roof sheathing						Number
	Wood shingles on wood strips‡			Asphalt shingles§ or roll roofing on wood sheathing‖			
	No roof insulation (rafters exposed) (U_r = 0.48)	½-in. insulating board on underside of rafters (U_r = 0.22)	1-in. insulating board on underside of rafters (U_r = 0.16)	No roof insulation (rafters exposed) (U_r = 0.53)	½-in. insulating board on underside of rafters (U_r = 0.23)	1-in. insulating board on underside of rafters (U_r = 0.17)	
	A	B	C	D	E	F	
0.10	0.085	0.073	0.066	0.087	0.074	0.067	19
0.11	0.093	0.078	0.07	0.094	0.079	0.071	20
0.12	0.099	0.082	0.074	0.10	0.083	0.075	21
0.13	0.10	0.087	0.077	0.11	0.088	0.079	22
0.14	0.11	0.092	0.080	0.11	0.093	0.083	23
0.15	0.12	0.096	0.084	0.12	0.097	0.086	24
0.16	0.12	0.10	0.087	0.13	0.10	0.089	25
0.17	0.13	0.10	0.090	0.13	0.10	0.092	26
0.18	0.13	0.11	0.093	0.14	0.11	0.095	27
0.19	0.14	0.11	0.096	0.15	0.11	0.098	28
0.20	0.15	0.11	0.098	0.15	0.12	0.10	29
0.21	0.15	0.12	0.10	0.16	0.12	0.10	30
0.22	0.16	0.12	0.10	0.17	0.12	0.11	31
0.23	0.17	0.12	0.10	0.17	0.12	0.11	32
0.24	0.17	0.13	0.11	0.18	0.12	0.11	33
0.25	0.17	0.13	0.11	0.18	0.13	0.11	34
0.26	0.18	0.13	0.11	0.19	0.13	0.11	35
0.27	0.18	0.13	0.11	0.19	0.13	0.12	36
0.28	0.19	0.14	0.11	0.19	0.14	0.12	37
0.29	0.19	0.14	0.12	0.20	0.14	0.12	38
0.30	0.20	0.14	0.12	0.20	0.14	0.12	39
0.34	0.21	0.15	0.12	0.22	0.15	0.13	40
0.35	0.21	0.15	0.12	0.22	0.15	0.13	41
0.36	0.22	0.15	0.12	0.23	0.15	0.13	42
0.37	0.22	0.15	0.13	0.23	0.16	0.13	43
0.45	0.25	0.17	0.13	0.26	0.17	0.14	44
0.59	0.29	0.18	0.14	0.30	0.19	0.15	45
0.61	0.29	0.18	0.14	0.31	0.19	0.15	46
0.62	0.30	0.18	0.15	0.31	0.19	0.15	47
0.67	0.31	0.19	0.15	0.33	0.20	0.16	48
0.69	0.31	0.19	0.15	0.33	0.20	0.16	49

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* Calculations based on ⅓ pitch roof ($n = 1.2$) using the following formula:

$$U + \frac{U_r \times U_{ce}}{U_r + \frac{U_{ce}}{n}}$$

U = combined coefficient to be used with ceiling area.
U_r = coefficient of transmission of the roof.
U_{ce} = coefficient of transmission of the ceiling.
n = the ratio of the area of the roof to the area of the ceiling.

† Use ceiling area (not roof area) with these coefficients.

‡ Based on 1 × 4 in. strips spaced 2 in. apart.

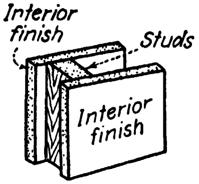
§ Coefficients in Columns D, E, and F may be used with sufficient accuracy for tile, slate, and rigid asbestos shingles on wood sheathing.

‖ Sheating assumed $^{25}\!/_{32}$ in. thick.

** Values of U_{ce} to be used in this column may be selected from Table 10.

TABLE 15.—COEFFICIENTS OF TRANSMISSION U OF FRAME PARTITIONS OR INTERIOR WALLS*

(Coefficients are expressed in B.t.u. per hr. per sq. ft. per deg. F. difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides.)

	Single partition (finish on one side only of studs)	Double partition (finish on both sides of studs)		Partition number
		No insulation between studs	1-in. blanket† between studs; 1 air space	
	A	B	C	
Metal lath and plaster‡	0.69	0.39	0.16	1
Gypsum board ($\frac{3}{8}$ in.), decorated	0.67	0.37	0.16	2
Wood lath and plaster	0.62	0.34	0.15	3
Gypsum lath ($\frac{3}{8}$ in.), plastered§	0.61	0.34	0.15	4
Plywood ($\frac{3}{8}$ in.), plain or decorated	0.59	0.33	0.15	5
Insulating board ($\frac{1}{2}$ in.), plain or decorated	0.36	0.19	0.11	6
Insulating-board lath ($\frac{1}{2}$ in.), plastered§	0.35	0.18	0.11	7
Insulating-board lath (1 in.), plastered§	0.19	0.12	0.082	8

* Coefficients not weighted; effect of studding neglected.

† For partitions with other insulations between studs refer to Table 5, using values in column B of above table in left-hand column of Table 5. *Example:* What is the coefficient of transmission *U* of a partition consisting of gypsum lath and plaster on both sides of studs with 2-in. blanket between studs? *Solution:* According to above table, this partition with no insulation between studs (No. 4B) has a coefficient of 0.34. Referring to Table 5, it will be found that a wall having a coefficient of 0.34 with no insulation between studs, will have a coefficient of 0.11 with 2 in. of blanket insulation between studs (No. 56B).

‡ Plaster assumed $\frac{3}{4}$ in. thick.

§ Plaster assumed $\frac{1}{2}$ in. thick.

TABLE 16.—COEFFICIENTS OF TRANSMISSION *U* OF DOORS, WINDOWS, SKYLIGHTS, AND GLASS BLOCK WALLS
(Coefficients are based on a wind velocity of 15 m.p.h., and are expressed in B.t.u. per hr. per sq. ft. per deg. F. difference in temperature between the air inside and outside of the door, window, skylight, or wall)

		Single	Double	Triple
Section *A*. Windows and skylights	*U*	1.13*†	0.45*‡	0.281*‡

	Nominal thickness, in.	Actual thickness, in.	*U* Exposed door	*U*§ With glass storm door
Section *B*. Solid wood doors†,‖	1	$\frac{25}{32}$	0.69	0.42
	1¼	$1\frac{1}{16}$	0.59	0.38
	1½	$1\frac{5}{16}$	0.52	0.35
	1¾	1⅜	0.51	0.35
	2	1⅝	0.46	0.32
	2½	2⅛	0.38	0.28
	3	2⅝	0.33	0.25

	Description	*U* Still air both sides	*U* Still air inside, 15 m.p.h. outside
Section *C*. Hollow glass block walls	Smooth surface glass blocks 7¾ × 7¾ × 3⅞ in. thick...	0.40	0.49
	Ribbed surface glass blocks 7¾ × 7¾ × 3⅞ in. thick...	0.38	0.46

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* See "Heating, Ventilating and Air Conditioning," by Harding and Willard, revised edition, 1932.

† It is sufficiently accurate to use the same coefficient of transmission for doors containing thin wood panels as that of single panes of glass, namely, 1.13 B.t.u. per hr. per sq. ft. per degree difference between inside and outside air temperatures.

‡ Air spaces assumed to be ¾ in. or more in width.

§ These values may also be used with sufficient accuracy for wood storm doors. Neglect storm doors if loose and use values for exposed doors.

‖ Computed using $C = 1.15$ for wood; $f_i = 1.65$ and $f_o = 6.0$.

Piping

Graphical Symbols for Drawings

Heating:

1. High-pressure steam
2. Medium-pressure steam
3. Low-pressure steam
4. High-pressure return
5. Medium-pressure return
6. Low-pressure return
7. Boiler blowoff
8. Condensate or vacuum-pump discharge
9. Feed-water pump discharge
10. Make-up water
11. Air-relief line
12. Fuel-oil flow
13. Fuel-öil return
14. Fuel-oil tank vent
15. Compressed air
16. Hot-water heating supply
17. Hot-water heating return

Air Conditioning:

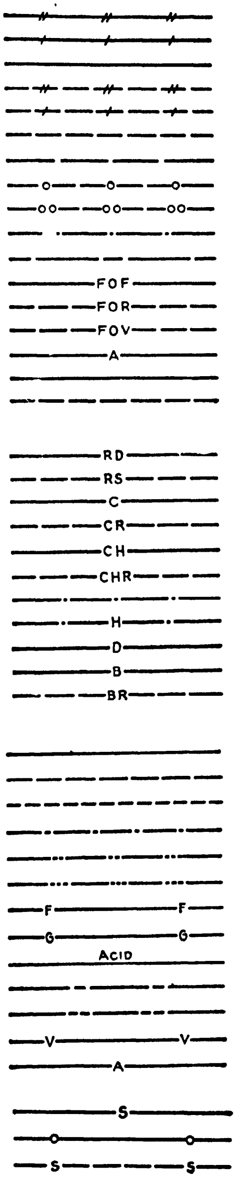
18. Refrigerant discharge
19. Refrigerant suction
20. Condenser-water flow
21. Condenser-water return
22. Circulating chilled- or hot-water flow
23. Circulating chilled- or hot-water terurn
24. Make-up water
25. Humidification line
26. Drain
27. Brine supply
28. Brine return

Plumbing:

29. Soil, waste, or leader (above grade)
30. Soil, waste, or leader (below grade)
31. Vent
32. Cold water
33. Hot water
34. Hot-water return
35. Fire line
36. Gas
37. Acid waste
38. Drinking-water flow
39. Drinking-water return
40. Vacuum cleaning
41. Compressed air

Sprinklers:

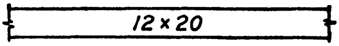
42. Main supplies
43. Branch and head
44. Drain



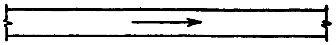
Ductwork

Graphical Symbols for Drawings

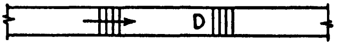
45. Duct (1st figure, width; 2d, depth)



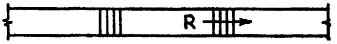
46. Direction of flow



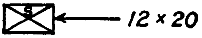
47. Inclined drop in respect to air flow



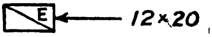
48. Inclined rise in respect to air flow



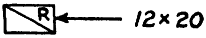
49. Supply-duct section



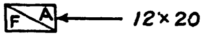
50. Exhaust-duct section



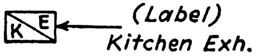
51. Recirculation-duct section



52. Fresh-air-duct section



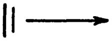
53. Other duct sections



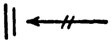
54. Register R

55. Grille G

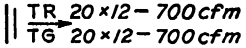
56. Supply outlet



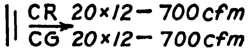
57. Exhaust inlet



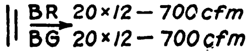
58. Top register or grille



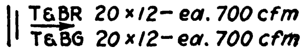
59. Center register or grille



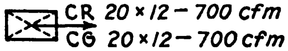
60. Bottom register or grille



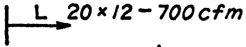
61. Top and bottom register or grille



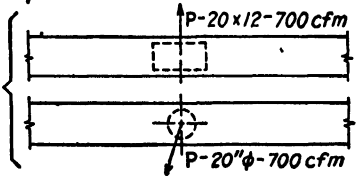
62. Ceiling register or grille



63. Louver opening



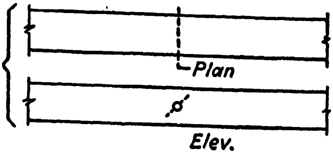
64. Adjustable plaque



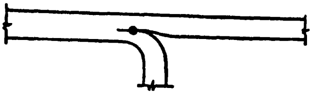
Ductwork

Graphical Symbols for Drawings

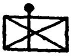
65. Volume damper



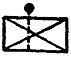
66. Deflecting damper



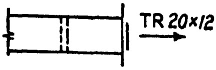
67. Deflecting damper, up



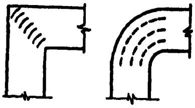
68. Deflecting damper, down



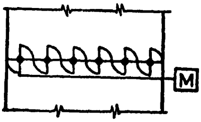
69. Adjustable blankoff



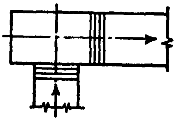
70. Turning vanes



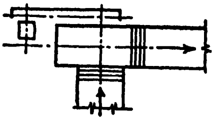
71. Automatic dampers



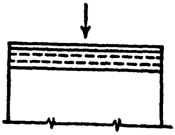
72. Canvas connections



73. Fan and motor with guard



74. Intake louvers and screen



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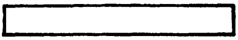
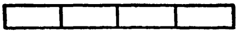
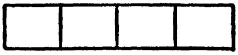
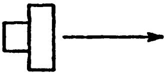
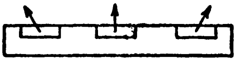
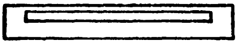
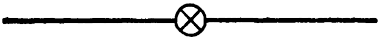
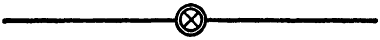
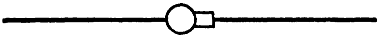
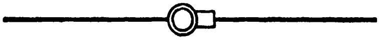
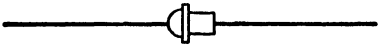
75. Heat-transfer surface, plan	
76. Wall radiator, plan	
77. Wall radiator on ceiling, plan	
78. Unit heater (propeller), plan	
79. Unit heater (centrifugal fan), plan	
80. Unit ventilator, plan	

Traps:

81. Thermostatic	
82. Blast thermostatic	
83 Float and thermostatic	
84. Float	
85. Boiler return	

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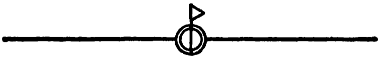
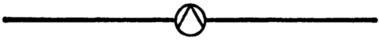
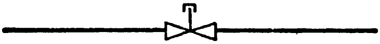
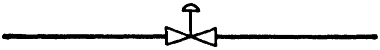
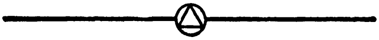
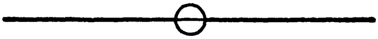
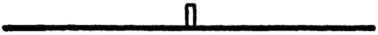
86. Reducing pressure	
87. Air line	
88. Lock and shield	
89. Diaphragm	
90. Air eliminator	
91. Strainer	
92. Thermometer	
93. Thermostat	

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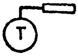
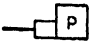
94. Thermostat (self-contained)

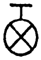
95. Thermostat (remote bulb)



96. Pressurestat

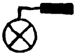


97. Hand expansion valve

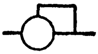


98. Automatic expansion valve

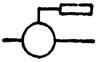
99. Thermostatic expansion valve



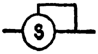
100. Evaporator pressure-regulating valve, throttling type



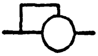
101. Evaporator pressure-regulating valve, thermostatic throttling type



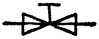
102. Evaporator pressure-regulating valve, snap-action valve



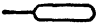
103. Compressor-suction pressure-limiting valve, throttling type



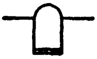
104. Hand shutoff valve



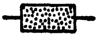
105. Thermal bulb



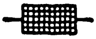
106. Scale trap



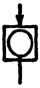
107. Dryer



108. Strainer



109. High side float



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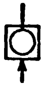
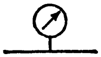
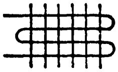
110. Low side float



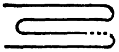
111. Gage



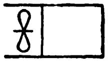
112. Finned-type cooling unit, natural convection



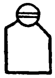
113. Pipe coil



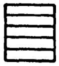
114. Forced-convection cooling unit



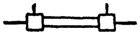
115. Immersion cooling unit



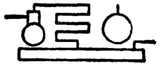
116. Ice-making unit



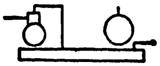
117. Heat interchanger



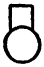
118. Condensing unit, air-cooled



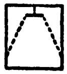
119. Condensing unit, water-cooled



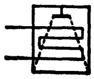
120. Compressor



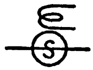
121. Cooling tower



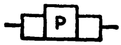
122. Evaporative condenser



123. Solenoid valve



124. Pressurestat with high-pressure cutout



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